



Calhoun: The NPS Institutional Archive
DSpace Repository

Theses and Dissertations

1. Thesis and Dissertation Collection, all items

1983-06

An analysis of smooth and axially finned,
rotating heat pipe condensers.

Kleinholz, Adam F.

Monterey, California. Naval Postgraduate School

<http://hdl.handle.net/10945/19635>

This publication is a work of the U.S. Government as defined in Title 17, United States Code, Section 101. Copyright protection is not available for this work in the United States.

Downloaded from NPS Archive: Calhoun



<http://www.nps.edu/library>

Calhoun is the Naval Postgraduate School's public access digital repository for research materials and institutional publications created by the NPS community. Calhoun is named for Professor of Mathematics Guy K. Calhoun, NPS's first appointed -- and published -- scholarly author.

Dudley Knox Library / Naval Postgraduate School
411 Dyer Road / 1 University Circle
Monterey, California USA 93943

Dudley Knox Library, NPS
Monterey, CA 93943

NAVAL POSTGRADUATE SCHOOL

Monterey, California



THESIS

AN ANALYSIS OF SMOOTH AND AXIALLY FINNED,
ROTATING HEAT PIPE CONDENSERS

by

Adam F. Kleinholz

June 1983

Thesis Advisor:

P. J. Marto

Approved for public release, distribution unlimited.

T208829

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) An Analysis of Smooth and Axially Finned, Rotating Heat Pipe Condensers		5. TYPE OF REPORT & PERIOD COVERED Master's Thesis; June 1983
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) Adam F. Kleinholz		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS Naval Postgraduate School Monterey, California 93940		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Postgraduate School Monterey, California 93940		12. REPORT DATE June 1983
		13. NUMBER OF PAGES 176
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)		15. SECURITY CLASS. (of this report) Unclassified
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE
16. DISTRIBUTION STATEMENT (of this Report) Approved for public release, distribution unlimited.		
17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)		
18. SUPPLEMENTARY NOTES		
19. KEY WORDS (Continue on reverse side if necessary and identify by block number) Heat Pipe Rotating Heat Pipe		
20. ABSTRACT (Continue on reverse side if necessary and identify by block number) A mathematical model is developed to determine the heat transfer rate of a cylindrical condenser section of a rotating heat pipe. This model is coupled to an existing code and an analysis is accomplished on both a smooth and axially finned condenser. The results of this analysis are compared to those of a similar analysis performed on a tapered condenser heat pipe using identical geometric and operating parameters.		

Results of the comparison indicate cylindrical condensers have a lower heat transfer rate than equivalent tapered condensers. This reduction in heat transfer rate is due to a greater condensate film thickness and is most significant in a smooth condenser.

Axially finned condensers with triangular and rectangular fin profiles are also compared. The rectangular fins are assumed to have adiabatic tips. Results indicate the heat transfer rates for these two profiles vary by only 0.40 per cent for both tapered and cylindrical condensers.

Approved for public release, distribution unlimited.

An Analysis of Smooth and Axially Finned,
Rotating Heat Pipe Condensers

by

Adam F. Kleinholz
Lieutenant, United States Navy
B.S., University of Oklahoma, 1975

Submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

from the

NAVAL POSTGRADUATE SCHOOL
June 1983

ABSTRACT

A mathematical model is developed to determine the heat transfer rate of a cylindrical condenser section of a rotating heat pipe. This model is coupled to an existing code and an analysis is accomplished on both a smooth and axially finned condenser. The results of this analysis are compared to those of a similar analysis performed on a tapered condenser heat pipe using identical geometric and operating parameters.

Results of the comparison indicate cylindrical condensers have a lower heat transfer rate than equivalent tapered condensers. This reduction in heat transfer rate is due to a greater condensate film thickness and is most significant in a smooth condenser.

Axially finned condensers with triangular and rectangular fin profiles are also compared. The rectangular fins are assumed to have adiabatic tips. Results indicate the heat transfer rates for these two profiles vary by only 0.40 per cent for both tapered and cylindrical condenser.

TABLE OF CONTENTS

I.	INTRODUCTION-----	15
A.	THE ROTATING HEAT PIPE-----	15
B.	BACKGROUND-----	17
C.	THESIS OBJECTIVES-----	20
II.	THEORETICAL ANALYSIS FOR A CYLINDRICAL HEAT PIPE----	21
A.	INTRODUCTION-----	21
B.	THEORY FOR A CYLINDRICAL SMOOTH CONDENSER-----	22
1.	Assumptions-----	22
2.	Condensate Momentum Equation (X-Direction)--	23
3.	Condensate Momentum Equation (Y-Direction)--	23
4.	Fluid Velocity-----	25
5.	Continuity Equation-----	26
6.	Energy Equation-----	27
7.	Determination of Heat Transfer Rate-----	29
C.	THEORY FOR A CYLINDRICAL AXIALLY FINNED CONDENSER-----	30
1.	Assumptions-----	30
2.	Mass Flow in the X-Direction-----	34
3.	Mass Flow in the Z-Direction-----	35
4.	Energy Equation for the Trough Condensate---	37
5.	Energy Equation for the Fin Condensate-----	38
6.	Continuity Equation-----	39
7.	Determination of the Heat Transfer Rate-----	40

III.	COMPUTER CODE DESCRIPTION-----	42
A.	GENERAL DESCRIPTION OF CODE-----	42
B.	INTERNALLY FINNED TAPERED CONDENSER SOLUTION---	44
C.	SMOOTH TAPERED CONDENSER SOLUTION-----	49
D.	SMOOTH CYLINDRICAL CONDENSER SOLUTION-----	52
E.	FINNED CYLINDRICAL CONDENSER SOLUTION-----	56
IV.	RESULTS AND DISCUSSIONS-----	60
V.	CONCLUSIONS AND RECOMMENDATIONS-----	93
A.	CONCLUSIONS-----	93
B.	RECOMMENDATIONS-----	93
APPENDIX A:	FILM PROFILE FINITE ELEMENT SOLUTION-----	95
A.	SMOOTH CONDENSER-----	95
B.	AXIALLY FINNED CONDENSER-----	103
APPENDIX B:	USER'S MANUAL-----	107
APPENDIX C:	SOURCE CODE LISTING-----	117
LIST OF REFERENCES	-----	174
INITIAL DISTRIBUTION LIST	-----	176

LIST OF TABLES

I.	Condenser Geometric Parameters Held Constant During All Analyses-----	62
II.	Condenser Geometric Parameters Applied as Required-----	62
III.	Operating Parameter Matrix-----	63
IV.	List of Parameters Used in Rectangular/ Triangular Fin Profile Comparison-----	68
V.	Results of Rectangular/Triangular Fin Profile Comparison-----	68
VI.	Surface Temperature Distribution for Rectangular and Triangular Axially Finned Tapered Copper Condensers at Middle Increment of Condensers-----	70
VII.	Surface Temperature Distribution for Rectangular and Triangular Axially Finned Cylindrical Copper Condensers at Middle Increment of Condensers-----	71

LIST OF FIGURES

1.	Schematic Drawing of a Tapered Condenser Rotating Heat Pipe-----	16
2.	Axially Finned Condenser Geometry Showing Fins, Troughs, and Lines of Symmetry-----	18
3.	Cross Section of Infinitesimal Fluid Element on Cylindrical Condenser Internal Surface-----	24
4.	Trough Section of Axially Finned Condenser-----	32
5.	Axially Finned Condenser Symmetric Fin Section-----	33
6.	Cross Section of Infinitesimal Fluid Element on Fin Surface of Axially Finned Condenser-----	36
7.	Axially Finned Condenser Symmetric Section Subdivided into 25 Linear Triangular Finite Elements-----	46
8.	Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Copper Condensers at 700 RPM-----	73
9.	Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Copper Condensers at 1400 RPM-----	74
10.	Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Copper Condensers at 2800 RPM-----	75
11.	Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Stainless Steel Condensers at 700 RPM-----	76
12.	Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Stainless Steel Condensers at 1400 RPM-----	77
13.	Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Stainless Steel Condensers at 2800 RPM-----	78

14.	Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Copper Condensers at 700 RPM-----	79
15.	Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Copper Condensers at 1400 RPM-----	80
16.	Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Taperd Copper Condensers at 2800 RPM-----	81
17.	Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Stainless Steel Condensers at 700 RPM-----	82
18.	Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Stainless Steel Condensers at 1400 RPM-----	83
19.	Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Stainless Steel Condensers at 2800 RPM-----	84
20.	Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Copper Cylindrical Condensers at 700 RPM-----	85
21.	Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Copper Cylindrical Condensers at 1400 RPM-----	86
22.	Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Copper Cylindrical Condensers at 2800 RPM-----	87
23.	Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Stainless Steel Cylindrical Condensers at 700 RPM-----	88
24.	Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Stailness Steel Cylindrical Condensers at 1400 RPM-----	89
25.	Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Stainless Steel Cylindrical Condensers at 2800 RPM-----	90
26.	Heat Transfer Rate versus Saturation Temperature for Copper and Stainless Steel Smooth Cylindrical Condensers at 1400 RPM-----	91

27.	Heat Transfer Rate versus Saturation Temperature for Copper and Stainless Steel Axially Finned Cylindrical Condensers at 1400 RPM-----	92
-----	--	----

TABLE OF SYMBOLS

A	cross sectional area for flow (ft^2)
c_p	specific heat (Btu/lbm-F)
\dot{m}	differential mass flow rate (lbm/hr)
dq	differential heat transfer rate (Btu/hr)
dq''	differential heat flux (Btu/hr-ft^2)
g	acceleration of gravity (ft/hr^2)
h	convective heat transfer coefficient ($\text{Btu/hr-ft}^2\text{-F}$)
h_{ext}	external heat transfer coefficient ($\text{Btu/hr-ft}^2\text{-F}$)
h_{fg}	latent heat of vaporization (Btu/lbm)
\bar{h}_{fg}	corrected latent heat of vaporization (Btu/lbm)
k_f	thermal conductivity of the condensate film (Btu/hr-ft-F)
k_w	thermal conductivity of condenser wall (Btu/hr-ft-F)
ℓ	length of element (ft)
L	condenser length (ft)
\dot{m}	mass flow rate (lbm/hr)
P	pressure (lb/ft^2)
P_v	pressure of vapor (lb/ft^2)
Q	heat transfer rate (Btu/hr)

r	internal radius of condenser (ft)
R	relaxation variable
T	temperature (degrees F)
T_{avg}	average fin surface temperature (degrees F)
T_{sat}	saturation temperature (degrees F)
T_w	condenser wall temperature (degrees F)
T_{fin}	fin surface temperature (degrees F)
T_{∞}	ambient temperature (degrees F)
thick	thickness of condenser wall (ft)
u	fluid velocity (ft/hr)
\bar{u}	average fluid velocity (ft/hr)
v	vapor velocity (ft/hr)
w	fluid velocity along fin surface in z-direction (ft/hr)
\bar{w}	average fluid velocity in z-direction (ft/hr)
x	coordinate measuring distance along the condenser length (ft)
y	coordinate measuring distance normal to condenser surface (ft)
z	coordinate measuring along surface of fin (ft)
z^*	distance along fin surface from fin tip to trough film thickness (ft)
GREEK	
α	fin half angle (degrees)
δ	film thickness along fin surface (ft)

δ^*	film thickness along condenser wall (ft)
ε	trough width (ft)
ω	angular velocity (radians/hr)
ϕ	condenser cone half angle (degrees)
ρ_f	density of liquid (lbm/ft ³)
τ	shear stress (lbf/ft ²)
τ_v	vapor liquid interface shear stress (lbf/ft ²)
μ	liquid dynamic viscosity (lbm/ft-hr)

ACKNOWLEDGMENT

The author wishes to express his sincere gratitude and appreciation to Dr. Paul J. Marto, Professor of Mechanical Engineering and Dr. David Salinas, Associate Professor of Mechanical Engineering for their unselfish assistance and guidance throughout the development of this thesis. The author feels he was most fortunate in that he had "the best of both worlds."

Finally, the author wishes to thank his wife, Nanci, who pointed out the light when there appeared to be nothing but darkness.

I. INTRODUCTION

A. THE ROTATING HEAT PIPE

The rotating, wickless heat pipe is a closed container designed to transfer large amounts of heat from rotating machinery components. Essentially, it consists of three main components: a cylindrical evaporator section, a condenser section which may be either tapered or cylindrical in shape, and a fixed amount of working fluid. A typical tapered rotating heat pipe is shown in Figure 1.

When the heat pipe is rotated about its longitudinal axis at a speed above a certain critical value, the working fluid forms an annulus in the evaporator section. Note in Figure 1 that the diameter of the evaporator is larger than the condenser. This larger diameter provides a greater liquid reservoir. As heat is added to the evaporator, the fluid in the evaporator will vaporize. The vapor will flow axially towards the condenser as a result of a slight pressure difference, transporting the latent heat of vaporization with it. In the condenser end, external cooling of the condenser causes the vapor to condense. In the case of a tapered heat pipe, the centrifugal force due to the rotation of the pipe has a component acting along the condenser wall which accelerates the liquid condensate back to the evaporator to complete the cycle.

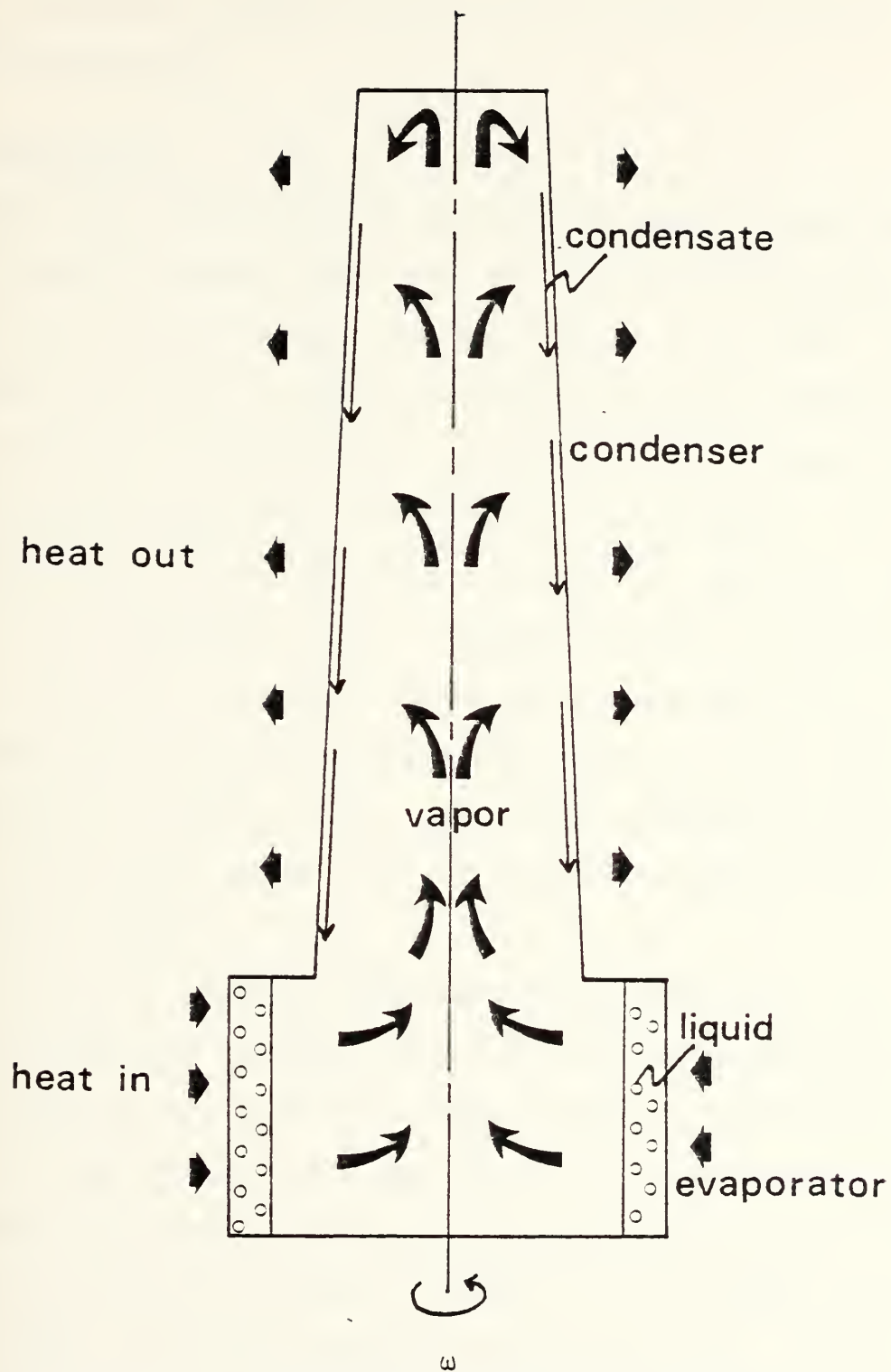


Figure 1. Schematic Drawing of a Tapered Condenser Rotating Heat Pipe

A cylindrical heat pipe, on the other hand relies on a hydrostatic pressure gradient to drive the liquid condensate back to the evaporator.

B. BACKGROUND

The first theoretical investigation into the performance of a tapered rotating heat pipe at the Naval Postgraduate School was accomplished by Ballback [Ref. 1] in 1969. He examined the limits in heat transfer controlled primarily by fluid dynamic considerations. In particular, he considered the following four limits on heat pipe performance: a) the boiling limit, b) the entrainment limit, c) the sonic limit and d) the condensing limit. Tantrakul [Ref. 2] calculated these limits for a specific heat pipe. He found the condensing limit was the controlling limitation. In fact, the calculated heat transfer rate, based on the condensing limit was 1/10th the heat transfer rate for the next lowest limit, the entrainment limit.

In order to overcome this condensing limitation and thus increase the heat transfer rate of the rotating heat pipe, the concept of an internally finned tapered rotating heat pipe was considered by Schafer [Ref. 3]. Schafer developed an analytical model for this tapered heat pipe with a triangular fin profile as shown in Figure 2. He assumed one dimensional heat conduction through the wall and fin. Corley [Ref. 4] developed a two-dimensional heat conduction model using a

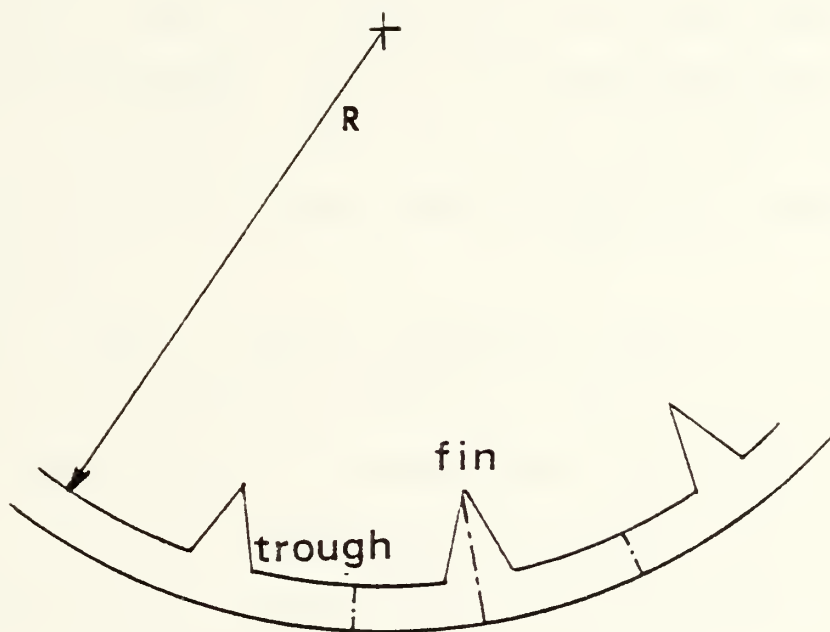


Figure 2. Axially Finned Condenser Geometry Showing Fins, Troughs, and Lines of Symmetry

Finite Element Method formulation for the same geometry. To overcome the problem of few nodal points along the fin surface, Corley assumed a parabolic temperature profile along the fin surface. Tantrakul [Ref. 2] modified Corley's computer code by increasing the number of finite elements from two to three in order to minimize the error at the apex of the fin. Purnomo [Ref. 5] developed a two-dimensional Finite Element Method solution to the steady state heat conduction problem using a linear triangular finite element. Davis [Ref. 6] modified Purnomo's code to make it compatible with COPES/CONMIN [Ref. 8], an optimization program. Davis, in his modification, found a coding error in Purnomo's [Ref. 5] code, that once corrected, permitted Purnomo's corrected code to converge to Schafer's [Ref. 3] results.

Purnomo's [Ref. 5] code is limited in that it is restricted to one particular condenser geometric configuration, namely: an axially finned tapered condenser heat pipe with a triangular fin profile. In that tapered finned condenser are difficult to manufacture, it is doubtful that any widespread practical application of this geometric configuration will result. Cylindrical condensers, on the other hand, can be manufactured with much less difficulty and might find practical application.

This being the case, a more practical and beneficial code would be one that could analyze cylindrical condenser rotating heat pipes, both finned and smooth. In actuality, the most beneficial code would be one that could analyze the following

four geometric configurations: 1) tapered-internally finned, 2) tapered-smooth, 3) cylindrical-internally finned, and 4) cylindrical-smooth. The theoretical heat transfer performance of the four geometries could then be compared to determine the advantages and disadvantages of each design. An additional advantage would be gained if different fin profiles, i.e., triangular vice rectangular, could also be analyzed and compared.

C. THESIS OBJECTIVES

The objectives of this thesis are:

- 1) Develop analytical models for both cylindrical-smooth and cylindrical-axially finned condensers.
- 2) Develop solution techniques to these analytical models that will account for temperature variations along the axial length of the condenser.
- 3) Modify Purnomo's [Ref. 5] code to provide a solution to the two-dimensional steady state conduction heat transfer problem for the following four geometric configurations:
 - a) tapered-smooth, b) tapered-finned, c) cylindrical-smooth, and d) cylindrical-finned.
- 4) Modify Purnomo's [Ref. 5] code to provide the additional capability of analyzing a rectangular fin profile with an adiabatic tip.
- 5) Obtain and compare results of the four geometric configurations given above for various operating conditions.

II. THEORETICAL ANALYSIS FOR A CYLINDRICAL HEAT PIPE

A. INTRODUCTION

In a cylindrical condenser heat pipe, the radius of the condenser is constant along the axial length of the condenser. The flow of the condensate in the absence of vapor-liquid interfacial shear, is dependent upon the variation in hydrostatic pressure with changes in film thickness along the surface of the heat pipe. Leppert and Nimmo [Refs. 8 and 9] investigated the phenomenon of film condensation on a flat horizontal plate. This situation is similar to film condensation on the inside surface of a rotating cylindrical condenser. In the case of a cylindrical condenser, the body force, rather than being the force of gravity, is now the centrifugal force caused by the rotation of the heat pipe. Weigenseil [Ref. 10] and Tantrakul [Ref. 2] compared experimental results for a cylindrical condenser rotating heat pipe with the theoretical results of Leppert and Nimmo [Refs. 8 and 9] and found good agreement. The Leppert and Nimmo solution was limited in that it was based on a constant surface temperature along the length of the plate. A rotating heat pipe, in actuality, has a temperature variation along the axial length of the condenser which in some cases, may be significant. This being the case, it was necessary to develop a mathematical model which would consider the axial temperature

variation in the solution of the heat transfer analysis. In the mathematical development that follows, a cylindrical smooth (unfinned) condenser will first be considered in that it is the simplest case. The model will then be extended to include a cylindrical axially finned condenser.

B. THEORY FOR A CYLINDRICAL SMOOTH CONDENSER

1. Assumptions

In developing the theoretical analysis, the following assumptions are made:

- a) Film condensation, not dropwise condensation occurs in the condenser.
- b) The condensate film undergoes laminar flow.
- c) Momentum changes through the condensate are small.
Thus, there is essentially a static balance of forces.
- d) The vapor exerts no drag in the condensate; there is no interfacial shear.
- e) The temperature distribution within the film is linear.
- f) The vapor space is essentially at one pressure, P_v .
- g) The density of the fluid is much greater than the density of the vapor. Thus, the density of the vapor can be neglected.
- h) The centrifugal force is much greater than the force of gravity and, thus, gravity may be neglected.
- i) Velocity gradients in the circumferential direction relative to the pipe wall are negligible.

- j) The condensate film thickness is much less than the radius of curvature of the condenser wall.
- k) The rotating heat pipe is operating at steady state conditions.

2. Condensate Momentum Equation (X-Direction)

By applying the above assumptions and the coordinate system shown in Figure 3, an analysis similar to Nusselt's original film condensation film theory may be used [Ref. 11]. Based on assumption c, a static force balance may be taken on an infinitesimal fluid element in the x-direction as shown in Figure 3. This force balance results in the following equation:

$$\Sigma F_x = 0 : \frac{\partial \tau}{\partial y} - \frac{\partial p}{\partial x} = 0 \quad (\text{eqn 2.1})$$

where τ = shear stress (lbf/ft²)

p = pressure (lbf/ft²)

x = co-ordinate measuring distance along surface (ft).

y = co-ordinate measuring distance normal to surface (ft).

3. Condensate Momentum Equation (Y-Direction)

In a similar manner, using Figure 3, a force balance in the y-direction yields:

$$\Sigma F_y = 0 : \frac{\partial p}{\partial y} + \rho_f \omega^2 r = 0 \quad (\text{eqn 2.2})$$

where ρ_f = density of the fluid (lbm/ft³)

ω = angular velocity (rad/hr)

r = radius (ft)

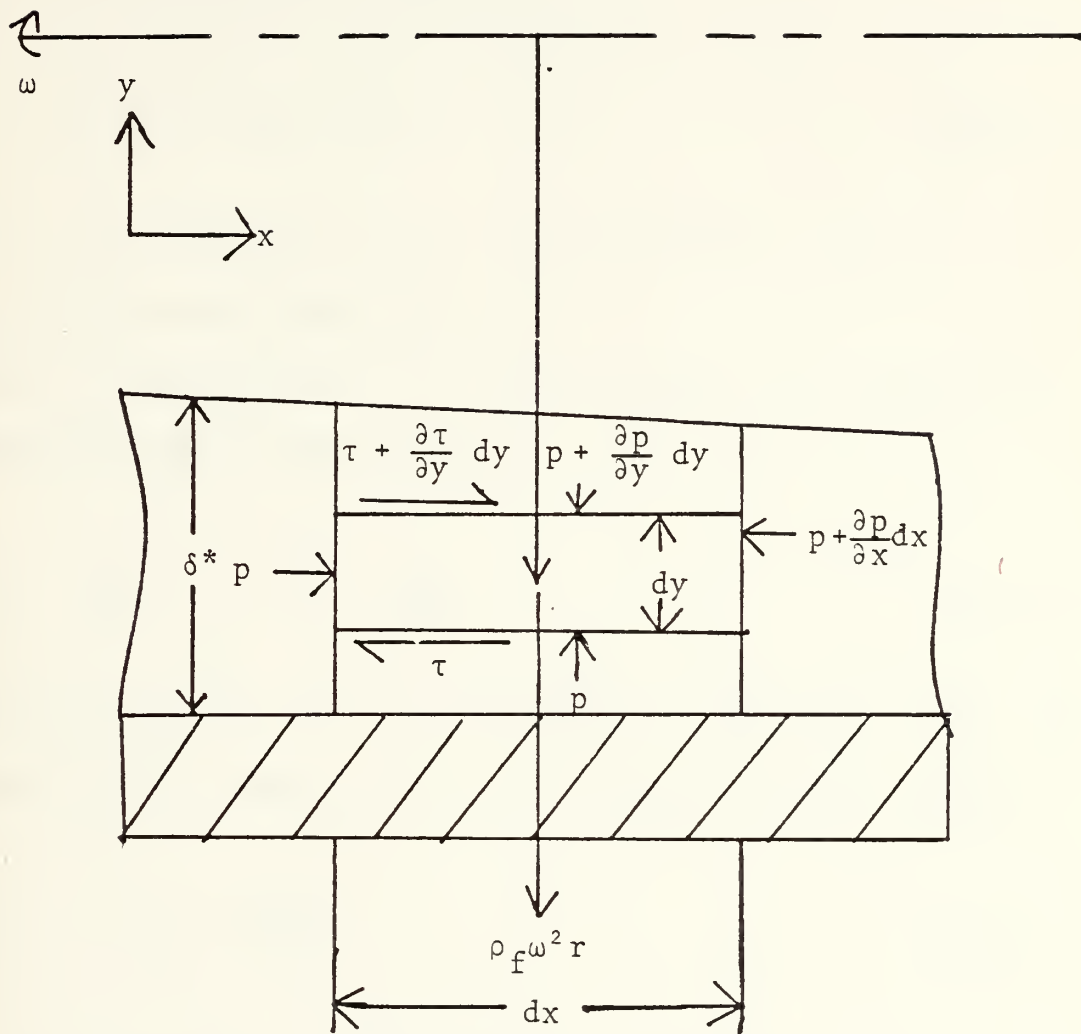


Figure 3. Cross Section of Infinitesimal Fluid Element on Cylindrical Condenser Internal Surface

4. Fluid Velocity

Integrating equation (2.2) between the limits y and δ^* for y and corresponding limits of P and P_v for pressure results in the following equation:

$$P = P_v + \rho_f \omega^2 r (\delta^* - y) \quad (\text{eqn 2.3})$$

where P_v = the pressure of the vapor (lbf/ft²), and

δ^* = film thickness (ft)

Differentiating equation (2.3) with respect to x yields the following expression for dP/dx :

$$\frac{dP}{dx} = \frac{dP_v}{dx} + \rho_f \omega^2 r \frac{d\delta^*}{dx} \quad (\text{eqn 2.4})$$

Applying assumption (f), (P_v is constant, therefore, $dP_v/dx=0$) and substituting equation (2.4) into equation (2.1) yields:

$$\frac{\partial \tau}{\partial y} = \rho_f \omega^2 r \frac{d\delta^*}{dx} \quad (\text{eqn 2.5})$$

Integrating equation (2.5) with the corresponding limits of integration y to δ^* and τ to 0 results in the following expression for shear stress:

$$\tau = \rho_f \omega^2 r \frac{d\delta^*}{dx} [y - \delta^*] \quad (\text{eqn 2.6})$$

But,

$$\tau = \mu \frac{\partial u}{\partial y} \quad (\text{eqn 2.7})$$

where μ = fluid dynamic viscosity (lbm/ft-hr)

u = condensate velocity (ft/hr)

Substituting equation (2.7) into equation (2.6) and integrating with the corresponding limits of integration 0 to y and 0 to u yields:

$$u = \frac{\rho_f \omega^2 r}{\mu} \frac{d\delta^*}{dx} \left[\frac{y^2}{2} - y\delta^* \right] \quad (\text{eqn 2.8})$$

The average velocity of the condensate may be found in the following manner:

$$\bar{u} = \frac{1}{\delta^*} \int_0^{\delta^*} u dy = \frac{1}{\delta^*} \int_0^{\delta^*} \frac{\rho_f \omega^2 r}{\mu} \frac{d\delta^*}{dx} \left[\frac{y^2}{2} - y\delta^* \right] dy \quad (\text{eqn 2.9})$$

or

$$\bar{u} = - \frac{\rho_f \omega^2 r}{\mu} \frac{d\delta^*}{dx} \left[\frac{\delta^{*2}}{2} \right] \quad (\text{eqn 2.10})$$

5. Continuity Equation

The continuity equation for mass flow requires that:

$$\dot{m} = \rho_f \bar{u} A \quad (\text{eqn 2.11})$$

where \dot{m} = condensate mass flow rate (lbm/hr)

A = cross sectional area of the fluid (ft²)

This can also be written as

$$\dot{m} = \int_0^{\delta^*} \rho_f \bar{u} 2\pi r dy \quad (\text{eqn 2.12})$$

Substituting equation (2.10) into equation (2.12) and integrating yields:

$$\dot{m} = - \frac{2\pi\rho_f \omega^2 r^2}{\mu} \frac{d\delta^*}{dx} \frac{\delta^{*3}}{3} \quad (\text{eqn 2.13})$$

Differentiating this equation with respect to x yields:

$$\frac{d\dot{m}}{dx} = - \frac{2\pi\rho_f \omega^2 r^2}{\mu} \frac{d}{dx} \left[\frac{d\delta^*}{dx} \frac{\delta^{*3}}{3} \right] \quad (\text{eqn 2.14})$$

6. Energy Equation

Having applied assumption (e), if the film surface temperature is at the saturation temperature (T_{sat}) of the vapor and if the wall of the axial increment is at a given constant temperature (T_w), then the heat transfer by conduction of a fluid element of surface area dA is:

$$dq = \frac{k_f (T_{\text{sat}} - T_w) dA}{\delta^*} \quad (\text{eqn 2.15})$$

where dq = differential heat transfer rate (Btu/hr)

dA = $2\pi r dx$ (ft²)

k_f = thermal conductivity of the condensate film
(Btu/hr-ft-F)

T_{sat} = saturation temperature (degrees F)

T_w = inside condenser wall temperature (degrees F)

Considering the change of phase and defining \bar{h}_{fg} as the average enthalpy change of the vapor in condensing to a liquid and subcooling to the average liquid temperature of the film, then dq is also defined by:

$$dq = \bar{h}_{fg} d\dot{m} \quad (\text{eqn 2.16})$$

where h_{fg} = latent heat of vaporization (Btu/lbm)

c_p = specific heat (Btu/lbm R)

$\Delta T = (T_{sat} - T_w)$

$\bar{h}_{fg} = h_{fg} + 0.35 \cdot c_p \cdot \Delta T$

Rearranging equation (2.16) and substituting this equation into equation (2.15) yields:

$$\frac{d\dot{m}}{dx} = \frac{k_f(T_{sat} - T_w) 2\pi r}{\delta^*} \quad (\text{eqn 2.17})$$

Finally coupling the energy and continuity equations result in the following differential equation:

$$\delta^* \frac{d}{dx} \left[\frac{d\delta^*}{dx} \delta^{*3} \right] = - \frac{3k_f(T_{sat} - T_w)\mu}{\rho_f^2 \omega^2 r \bar{h}_{fg}} \quad (\text{eqn 2.18})$$

Equation (2.18) can be solved using the Finite Element Method to provide the film thickness profile along the axial

length of a cylindrical condenser. Appendix A provides a detailed description of this solution. Once the film profile is known, a steady state two-dimensional heat conduction analysis can be performed.

7. Determination of Heat Transfer Rate

Assume that the cylindrical condenser section of the rotating heat pipe is divided axially into a number of increments. Then for any axial increment of a cylindrical condenser, the differential heat flux can be determined by the following expression.

$$dq^{11} = \frac{(T_{sat} - T_{\infty})}{\frac{\delta}{k_f} + \frac{thick}{k_w} + \frac{1}{h_{ext}}} \quad (eqn 2.19)$$

where T_{∞} = ambient temperature (degrees F)

thick = thickness of the condenser wall (ft)

k_w = thermal conductivity of the wall material
(Btu/hr-ft-F)

h_{ext} = external heat transfer coefficient
(Btu/hr-ft²-F)

Note the three terms in the denominator are the thermal resistances of the film, wall and external convection respectively.

The differential heat transfer rate for any increment can be found by the following relationships:

$$dq = dq^{11} \cdot 2\pi r dx \quad (eqn 2.20)$$

or

$$dq = \frac{2\pi r(T_{sat} - T_w) dx}{\frac{\delta}{k_f} + \frac{thick}{k_w} + \frac{1}{h_{ext}}} \quad (eqn 2.21)$$

Equation (2.21) represents the total heat transfer rate for an incremental section of width dx . To find the total heat transfer rate for the entire cylindrical condenser, the incremental heat rates must be summed over the entire length of the condenser. Therefore:

$$Q_{total} = \sum_{i=1}^{NDIV} dq \quad (eqn 2.22)$$

where $NDIV$ = total number of axial increments.

C. THEORY FOR A CYLINDRICAL AXIALLY FINNED CONDENSER

1. Assumptions

Referring to Figure 4, it is obvious that the analysis of a cylindrical internally finned condenser is more complicated due to the mass flow from the fins into the trough region between the fins. For this reason, in addition to the simplifying assumptions made for the smooth condenser which are listed in the previous section, the following assumptions must also be made:

- a) Referring to Figure 5, the mass flow along the fin surface does not flow axially in the x -direction, but only

along the surface of the fin in the z-direction into the trough. Thus, mass flow in the axial direction is only permitted in the trough region between the fins. This is a reasonable assumption in that the film thickness along the fin surface is very small in relation to the film thickness in the trough. This being the case, the hydrostatic force in the x-direction on a fin fluid element will be much less than the centrifugal force component in the z-direction on that same fluid element forcing that fluid element into the trough.

- b) Just as in the axial direction, there is no pressure change along the surface of the fin in the z-direction.
- c) It will be assumed that the temperature along the convective surface of the fin is at a constant value (T_{avg}). This average fin surface temperature is the arithmetic average of the fin tip temperature and the fin base surface temperature where the fin intersects with the wall of the condenser. This is a valid assumption if the fin section is divided into a sufficient number of finite elements. Purnomo's [Ref. 5] results indicate a less than one degree variation, even for very large fin half angles. This variation in temperature will have an insignificant effect on film thickness along the surface of the fin and can be neglected by using an average value.

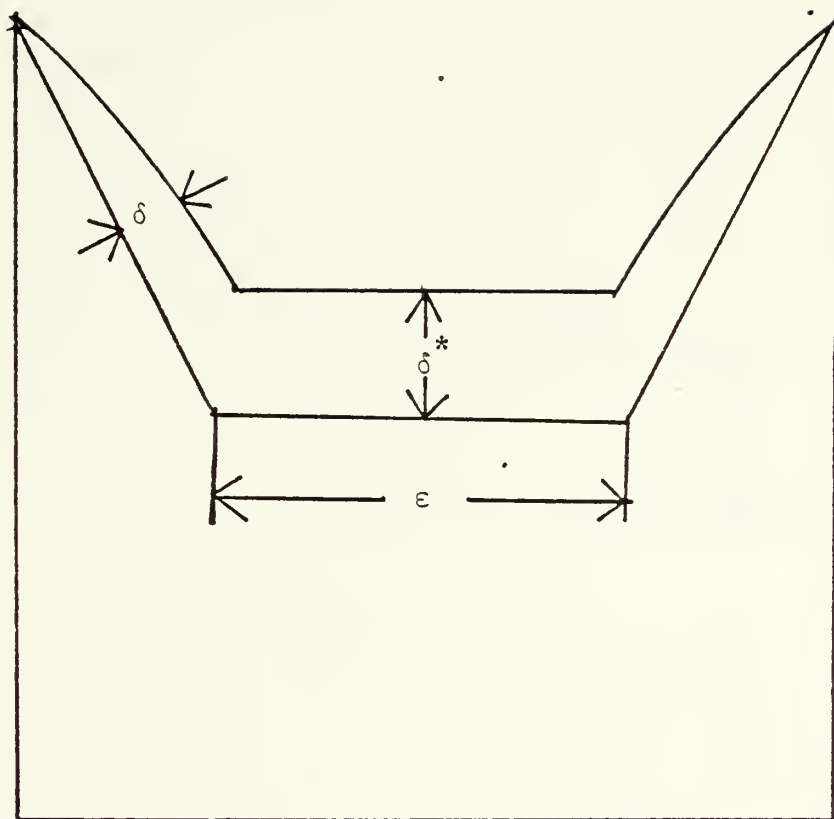


Figure 4. Trough Section of Axially Finned Condenser

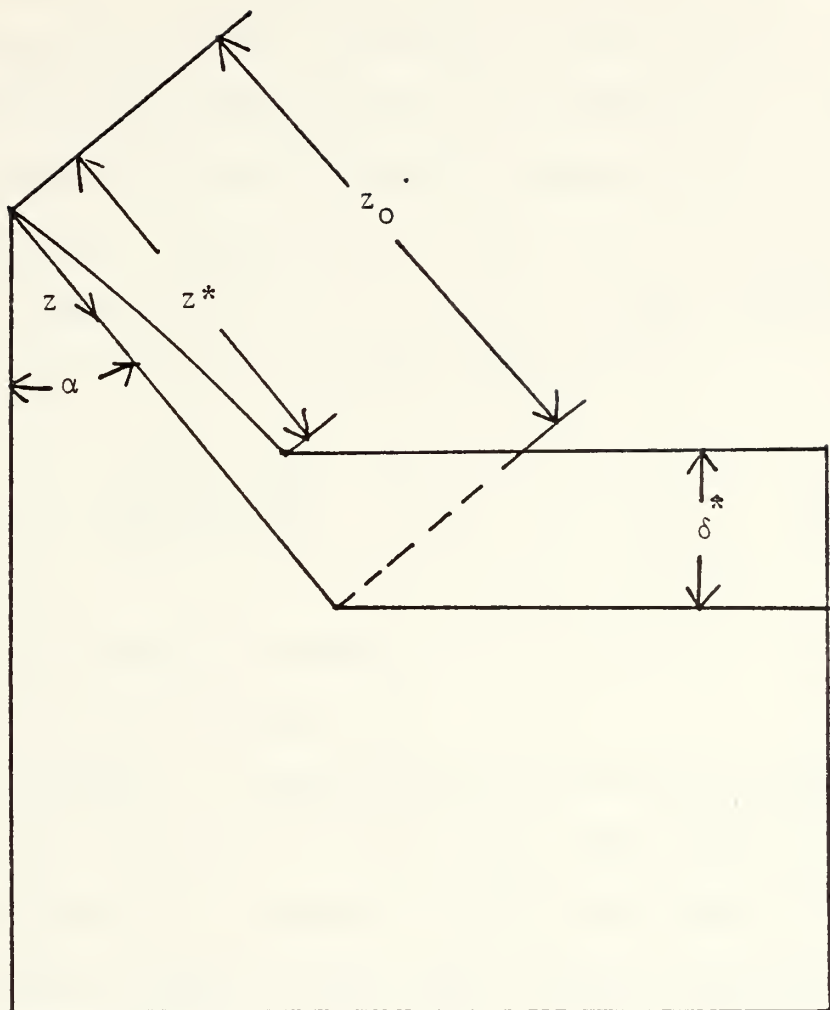


Figure 5. Axially Finned Condenser Symmetric Fin Section

2. Mass Flow in the X-Direction

As a result of assumption (a), the resulting momentum equation in both the x and y directions as well as the equations for velocity and mean velocity are identical to those developed in section B for the smooth condenser and will not be redeveloped here. Looking now at mass flow in the x-direction which is limited to flow only in the trough, the mass flow rate is given by the following expression:

$$\dot{m}_{\text{total}} = - \frac{\rho_f \omega^2 r}{3\mu} \frac{d\delta^*}{dx} \delta^{*2} (\epsilon \delta^* + \delta^{*2} \tan \alpha) \quad (\text{eqn 2.23})$$

where α = fin half angle (radians)

ϵ = width of the trough (ft)

Note that the quantity in parentheses is the cross sectional area of the film condensate in the trough (See Figure 4).

Taking the derivative of equation (2.23) with respect to x yields the rate of change of mass flow in the trough for a given axial increment.

$$\frac{d\dot{m}_{\text{total}}}{dx} = \frac{\rho_f \omega^2 r}{3\mu} \frac{d}{dx} \left[\frac{d\delta^*}{dx} (\epsilon \delta^{*3} + \delta^{*4} \tan \alpha) \right] \quad (\text{eqn 2.24})$$

Equation (2.24) represents the rate of change of the total mass flow rate with respect to x in the x-direction. This equation must be coupled with the energy equations for the fin and trough to develop a representation of the film profile in the trough.

3. Mass Flow in the Z-Direction

Examining an infinitesimal fluid element on the surface of a fin for any axial increment of width Δx , as shown in Figure 6, the momentum equation in the z-direction becomes:

$$\frac{\partial \tau_z}{\partial y} = \frac{\partial P}{\partial z} - \rho_f \omega^2 r \cos \alpha \quad (\text{eqn 2.25})$$

where τ_z = shear stress in the z-direction (lbf/ft²)

z = co-ordinate measuring distance along the surface
of the fin (ft)

Neglecting dP/dz based on assumption (b), and integrating equation (2.25) from τ_z to 0 and y to δ yields:

$$\tau_z = \mu \frac{\partial w}{\partial y} = \rho_f \omega^2 r \cos \alpha (\delta - y) \quad (\text{eqn 2.26})$$

where w = fluid velocity in the z-direction (ft/hr)

δ = fin film thickness along the surface of the fin (ft)

Note, δ , the fin film thickness should not be confused with δ^* , the film thickness in the trough. Integrating equation (2.26) from 0 to w and 0 to y provides the following expression for fluid velocity:

$$w = \frac{\rho_f \delta^2 r \cos \alpha}{\mu} \left(\delta(z)y - \frac{y^2}{2} \right) \quad (\text{eqn 2.27})$$

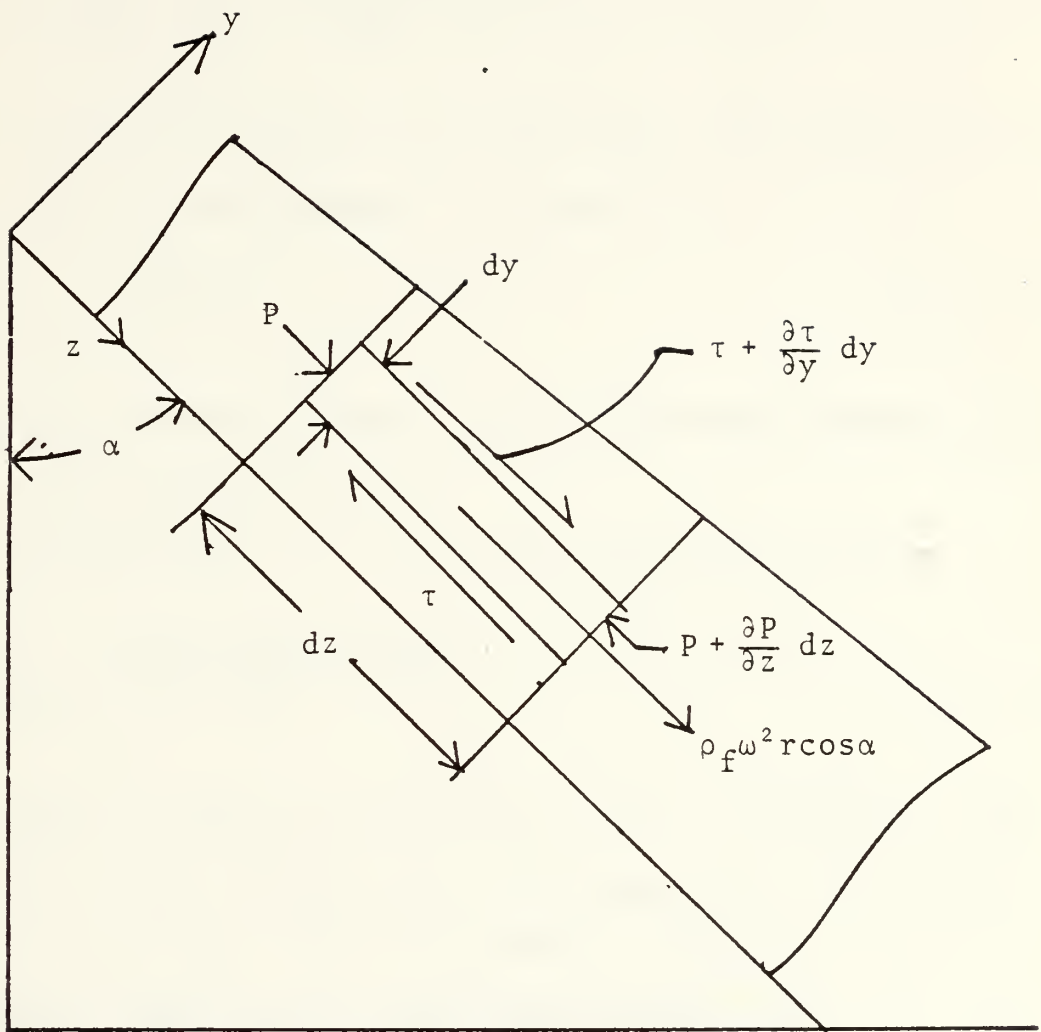


Figure 6. Cross Section of Infinitesimal Fluid Element on Fin Surface of Axially Finned Condenser

This relationship may be used to find the average fin fluid velocity \bar{w} :

$$\bar{w} = \frac{1}{\delta} \int_0^{\delta} w dy = \frac{\rho_f \omega^2 r \cos \alpha \delta(z)^2}{3\mu} \quad (\text{eqn 2.28})$$

The mass flow rate in the z-direction along the surface of the fin for a given axial increment is given by:

$$\dot{m}_{\text{fin}} = \rho_f \bar{w} dA \quad (\text{eqn (2.29)})$$

where dA = the cross sectional area of the fluid flowing along the fin surface (ft^2)

Substituting equation (2.28) into equation (2.29) yields:

$$\dot{m}_{\text{fin}} = \frac{\rho_f^2 \omega^2 r \cos \alpha \delta^3(z) dx}{3\mu} \quad (\text{eqn 2.30})$$

This equation is identical to equation (15) of Schafer's [Ref. 3] analysis if the condenser cone half angle (\emptyset) is set equal to 0 which is the case for a cylindrical condenser.

4. Energy Equation for the Trough Condensate

An energy balance on an infinitesimal fluid element in the trough of an axial increment of width dx with surface area $\epsilon \cdot dx$ yields the following expression for heat transfer by conduction:

$$dq_{\text{trough}} = \frac{k_f (T_{\text{sat}} - T_w) \epsilon dx}{\delta^*} \quad (\text{eqn 2.31})$$

Note also, that the trough heat transfer rate is given by:

$$dq_{\text{trough}} = \bar{h}_{fg} d\dot{m}_{\text{trough}} \quad (\text{eqn 2.32})$$

Combining equations (2.31) and (2.32) and dividing by dx results in the following:

$$\frac{d\dot{m}_{\text{trough}}}{dx} = \frac{k_f(T_{\text{sat}} - T_w)\epsilon}{\bar{h}_{fg} \delta^*} \quad (\text{eqn 2.33})$$

Equation (2.33) is an expression for incremental change in mass flow rate with respect to x due to condensation in the trough region.

5. Energy Equation for the Fin Condensate

An energy balance on a differential element of surface area $dx \cdot dz$ yields the following relationship for differential heat into the fin:

$$dq_{\text{fin}} = h_{fg} d\dot{m}_{\text{fin}} = \frac{k_f(T_{\text{sat}} - T_{\text{fin}}(z))dx dz}{\delta(z)} \quad (\text{eqn 2.34})$$

where $T_{\text{fin}}(z)$ = fin surface temperature at some position z
along the surface of the fin (degrees F)

Since the fin condensate mass is assumed to flow only in the z -direction, equation (2.30) is differentiated with respect to z and substituted into equation (2.34). After substitution and rearrangement, the following equation results:

$$\delta(z)^3 d\delta(z) = \frac{k_f(T_{sat} - T_{fin}(z))dz}{\rho_f^2 \omega^2 r h_{fg} \cos\alpha} \quad (\text{eqn 2.35})$$

Applying assumption (c) i.e., $T_{fin}(z)$ equals T_{avg} for all z and integrating equation (2.35) from 0 to δ and 0 to z yields the following relationship for fin film thickness $\delta(z)$:

$$\delta(z) = \left[\frac{4 k_f(T_{sat} - T_{avg})\mu z}{\rho_f^2 \omega^2 r h_{fg} \cos\alpha} \right]^{1/4} \quad (\text{eqn 2.36})$$

where T_{avg} = average fin surface temperature (degrees F). Substituting equation (2.36) into equation (2.30) and solving for rate of change of mass flow rate of the fin with respect to x for an increment of width dx yields:

$$\frac{d\dot{m}_{fin}}{dx} = \frac{2\rho_f^2 \omega^2 r \cos\alpha}{3\mu} \left[\frac{4 k_f(T_{sat} - T_{avg})\mu z^*}{\rho_f^2 \omega^2 r h_{fg} \cos\alpha} \right]^{1/4} \quad (\text{eqn 2.37})$$

where $z^* = z - \delta^*/\cos(\alpha)$. Note, z^* is the distance along the surface of the fin from the fin tip to the trough film thickness (δ^*). Note also, that the right hand side of equation (2.37) is multiplied by two; this accounts for mass flow from the fins on both sides of the trough.

6. Continuity Equation

For any axial increment of length dx , continuity dictates that:

$$\frac{d\dot{m}_{total}}{dx} = \frac{d\dot{m}_{fin}}{dx} + \frac{d\dot{m}_{trough}}{dx} \quad (\text{eqn 2.38})$$

Substituting equations (2.24), (2.33) and (2.37) into equation (2.38) and rearranging yields:

$$\delta^* \frac{d}{dx} \left[\frac{d\delta^*}{dx} (\epsilon \delta^{*3} + \delta^{*4} \tan \alpha) \right] = - \frac{3 k_f (T_{sat} - T_w) \mu \epsilon}{\rho_f^2 \omega^2 r \bar{h}_{fg}}$$

$$- 2\delta^* \cos \alpha \left[\frac{4 k_f (T_{sat} - T_{avg}) \mu z^*}{\rho_f^2 \omega^2 r \bar{h}_{fg} \cos \alpha} \right]^{3/4} \quad (\text{eqn 2.39})$$

Equation (2.39) can be solved using the Finite Element Method formulation provided in Appendix A. The solution of this equation provides the film thickness profile along the axial length of a cylindrical finned condenser.

7. Determination of the Heat Transfer Rate

Once the film profile has been determined within the trough, the local convective heat transfer coefficient can be found for the trough using the following relationship:

$$h(x)_{trough} = \frac{k_f}{\delta^*(x)} \quad (\text{eqn 2.40})$$

In a similar manner, the local heat transfer coefficient along the surface of the fin can be found by:

$$h(z)_{fin} = \frac{k_f}{\delta(z)} \quad (\text{eqn 2.41})$$

The differential heat transfer rate for any fin section, as shown in Figure 4, of axial incremental length dx is:

a) for the trough:

$$dq_{\text{trough}} = \frac{(T_{\text{sat}} - T_w) \epsilon dx}{h(x)_{\text{trough}}} \quad (\text{eqn 2.42})$$

where $\epsilon \cdot dx$ is the surface area of the trough, and

b) for the fin surface:

$$dq_{\text{fin}} = 2 \int_0^{z_0} \frac{(T_{\text{sat}} - T_{\text{avg}}) dx dz}{h(z)_{\text{fin}}} \quad (\text{eqn 2.43})$$

where z_0 is the surface length of the fin.

The total differential heat transfer rate per axial increment is found by summing equation (2.42) and (2.43) for the total number of fins. That is:

$$dq_{\text{total}} = \sum_1^{\text{NFIN}} (dq_{\text{fin}} + dq_{\text{trough}}) \quad (\text{eqn 2.44})$$

where NFIN is the total number of axial fins.

In a similar manner, the total heat transfer rate for the entire finned condenser can be found by the following relationship:

$$Q_{\text{total}} = \sum_1^{\text{NDIV}} dq_{\text{total}} \quad (\text{eqn 2.45})$$

where NDIV is the total number of axial increments.

III. COMPUTER CODE DESCRIPTION

A. GENERAL DESCRIPTION OF CODE

The computer code consists of a main body and eight subroutines. Basically, the code which is provided in Appendix C is a modification of Purnomo's [Ref. 5] code. The function of each subroutine used in the code is as follows:

- a) "CORRES" established the correspondence between the local and global nodal points used in the Finite Element Method solution for the two-dimensional steady state heat conduction problem. In so doing, "CORRES" also numbers all elements and nodal points in the finite element model and assigns local nodal points to each of the elements. In addition "CORRES" also defines major element numbers used in other subroutines as control parameters.
- b) "COORD" defines the x and y coordinates for all nodal points in the finite element heat conduction problem model.
- c) "DLSTAR" determines the film thickness (δ^*) on the surface of a smooth condenser or in the trough in a finned condenser.
- d) "HTCOEF" determines the heat transfer coefficient for all convective surface elements.

- e) "FORMAF" formulates the Finite Element Method equations for the two-dimensional steady state heat conduction problem.
- f) "BANDEC" is an equation solver for a symmetric matrix which has been transformed into banded form. "BANDEC" will return the solution to the two-dimensional heat conduction problem.
- g). "HTCALC" determines the elemental, incremental and total heat transfer rates.
- h) "DELCRV" determines the condensate film profile in a cylindrical condenser.

Two additional Naval Postgraduate School computer library routines are also used in the code:

- a) "DPOLRT" is a nonIMSL double precision library routine that determines the roots of a real polynomial. This routine is called by "DLSTAR" to determine the film thickness for the succeeding increment in the analysis of a tapered condenser.
- b) "LEQT2F" is an IMSL double precision library routine that solves a set of simultaneous linear equations. This routine is called by "DELCRV" to solve the Finite Element Method equations for the cylindrical condenser film profile problem. The resulting film profile is then used in the heat conduction analysis.

In order to use the computer code to analyze heat transfer in a rotating heat pipe, nine data cards are required. A user's

guide describing these data cards and required input is provided in Appendix B. The input data, describing the geometric configuration of the rotating heat pipe as well as the operating parameters determines which solution technique is utilized in the analysis. The solution technique for each of the four condenser geometries, i.e., tapered-smooth, tapered-axially finned, cylindrical-smooth and cylindrical-axially finned is different. In all cases however, the Finite Element Method is used to solve the two-dimensional steady state heat conduction problem. This solution is the one developed by Purnomo [Ref. 5] and has not been modified. Details of the development of this solution are described in detail in Purnomo's thesis [Ref. 5] and will not be repeated here. This being the case, each of the four solution techniques will now be discussed in detail.

B. INTERNALLY FINNED TAPERED CONDENSER SOLUTION

The complete development of this solution technique is provided in Purnomo's [Ref. 5] thesis and will not be redeveloped. When an equation is required for clarity, the equation in final form will be provided. Where there is a modification to Purnomo's [Ref. 5] code, this modification will be noted.

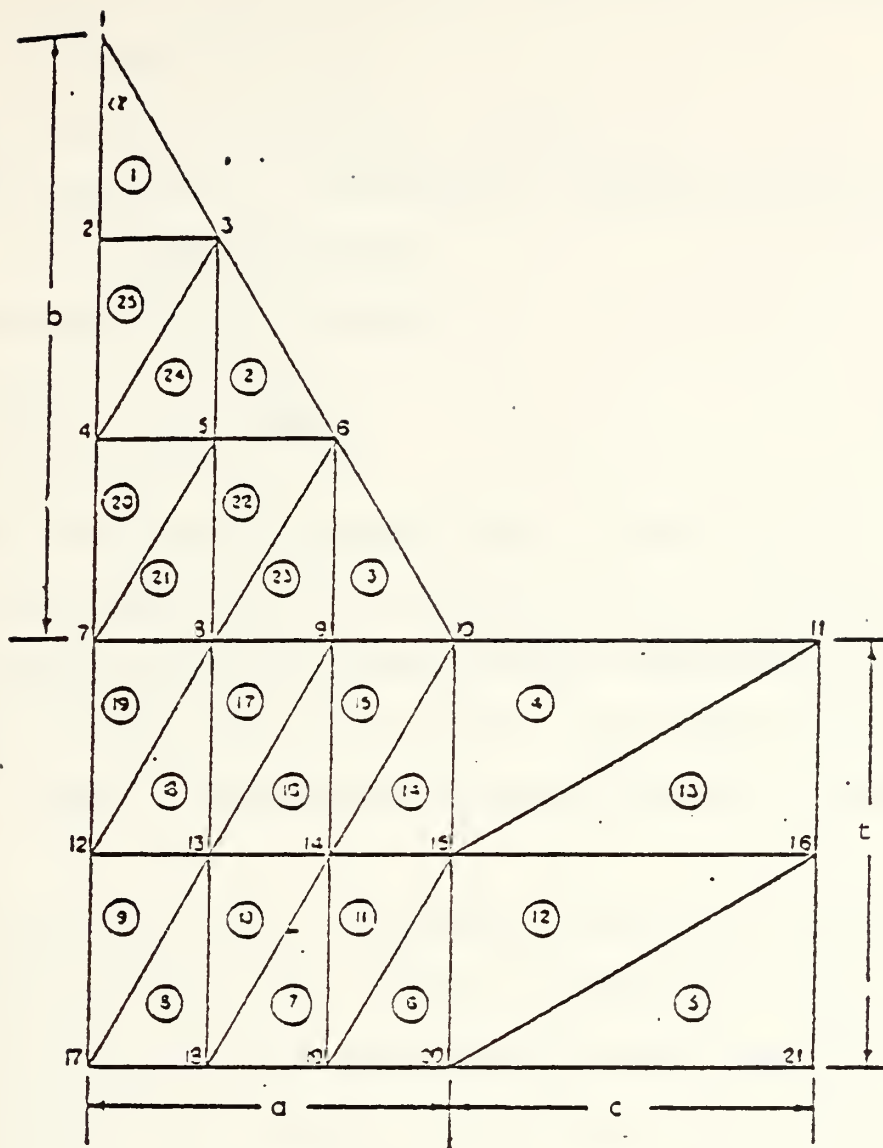
The condenser of the internally finned-tapered condenser is divided into NDIV axial increments. These axial increments are then subdivided circumferentially into ZFIN number of subincrements where ZFIN is the total number of fins.

These subincrements are then divided in half to form the basic symmetric unit, one-half of a fin-trough section as shown in Figure 1. This unit section is then divided into a number of linear triangular elements. The number of elements depend on the input parameters. The only limitation is that the number of system nodal points must not exceed 100; otherwise, certain variables, e.g., x and y, would exceed allotted storage values. Figure 7 shows a unit section subdivided into 25 elements. After this unit is subdivided, each nodal point is assigned an x and y coordinated based on the geometric input parameters.

To start the iteration, two initial values are required: 1) an initial temperature for the nodal points along the internal convective boundary, and 2) an initial trough film thickness (δ^*). The initial temperature is provided as an input parameter and the initial trough film thickness at the first increment is provided by a relationship taken from an analysis by Sparrow and Gregg for condensation on a rotating disk [Ref. 12].

Once these values are known, the heat transfer coefficient for the internal convective surface elements are found using the following relationships:

$$h(z)_{fin} = \frac{k_f}{\delta(z)} = \left[\frac{k_f \rho_f^2 \omega^2 (r+x \sin \theta) h_{fg} \cos \theta \cos \alpha}{4\mu (-AA \cdot z^3/3 - BB \cdot z^2/2 + (T_{sat} - T_1)z)} \right]^{1/4} \quad (\text{eqn 3.1})$$



$a = 0.02953$ inches

$b = 0.05906$ inches

$c = 0.04412$ inches

$t = 0.05118$ inches

Figure 7. Axially Finned Condenser Symmetric Section Subdivided into 25 Linear Triangular Finite Elements

where x = distance from condenser end ($x=0.0$) to midpoint
of increment (ft)

T_1 = fin apex temperature (degrees F)

AA, BB = constants in the parabolic temperature determined
by a Langrangian fit.

For the trough surface elements:

$$h(x)_{\text{trough}} = \frac{k_f}{\delta^*(x)} \quad (\text{eqn 3.2})$$

These heat transfer coefficients, along with the thermal conductivity and x and y coordinates of the nodal points are used to form the Finite Element Method equations for the two-dimensional heat conduction problem. The equations are then solved to yield a temperature distribution in the symmetric section.

The above iteration is repeated, where, now the solution temperature distribution from the previous iteration is used to calculate the heat transfer coefficients along the convective surface fin elements as well as a new δ^* using Sparrow's and Gregg's relationship [Ref. 12]. This new δ^* , in turn, is used to determine the heat transfer coefficients of the trough elements.

Again, the Finite Element Method subroutines will yield a temperature distribution for the symmetric section. At this point, the nodal point temperatures are checked for convergence using the following relationship:

$$\text{Max} \left| \frac{T_{i,j} - T_{i,j-1}}{T_{i,j}} \right| \leq \text{CRIT} \quad i = 1, 2, \dots, \text{NSNP} \quad (\text{eqn } 3.3)$$

where NSNP = number of system nodal points

j = present iteration

j-1 = previous iteration

CRIT = convergence criterion

If this convergence test is successful for all nodal points, the increment is considered solved. If any one nodal point fails, the iterative process is repeated until convergence is met. The convergence test in equation (3.3) is different than the one used by Purnomo [Ref. 5] in his thesis. Purnomo compared incremental heat transfer rates per unit of condenser length, Q_i , rather than temperature as is done in the modified code.

If convergence is met, the heat transfer rate is determined. From this heat transfer rate, the incremental mass flow rate is determined by the following equation:

$$\dot{m}_{\text{total}} = \frac{2Q_i \Delta x}{h_{fg}} \quad (\text{eqn } 3.4)$$

where Q_i = heat transfer rate per unit length (Btu/hr-ft)

Δx = incremental width (ft)

Using this value of incremental mass flow rate determined by equation (3.4), the following equation is used to calculate the subsequent interval's trough condensate film thickness (δ^*) with a polynomial rootfinder subroutine:

$$\dot{m}_{total} = \frac{\rho_f^2 \omega^2 (r+x \sin \theta) \delta^{*2}(x) \sin \theta (\delta^*(x) \epsilon + \delta^{*2}(x) \tan \alpha)}{3\mu} \quad (\text{eqn 3.5})$$

where ϵ = trough width (ft)

This resulting value of $\delta^*(x)$ is then defined as the trough film thickness for the next increment. In addition, the solution temperature distribution from the previous iteration is used as the starting temperature distribution for the next increment.

This iterative process at each increment is repeated until convergence is met, and, is continued at each increment until the entire length of the condenser has been transversed. Incremental heat rates are then summed to yield the total heat transfer rate. That is:

$$Q_{total} = 2 * ZFIN * \sum_{i=1}^{NDIV} Q_i \cdot \Delta x \quad (\text{eqn 3.6})$$

where ZFIN is the total number of axial fins.

Once the total heat transfer rate has been determined, the problem is solved and pertinent data is provided as output.

C. SMOOTH TAPERED CONDENSER SOLUTION

The heat pipe condenser is divided into NDIV number of axial increments as in the finned-tapered condenser solution. These axial increments are then subdivided into 360 segments of equal length; these segments are the basic symmetric unit.

This unit section, is divided into a number of linear triangular elements with the same limitation as before; the number of system nodal points must not exceed 100. The system nodal points are then assigned x and y coordinates based on the input geometric parameters.

To start the iterative process, as in the finned-tapered case, the initial value of temperature which is an input parameter is used to solve for the initial value of fin thickness (δ^*) based on the Sparrow and Gregg analysis [Ref. 12].

Once this initial value of δ^* is known, the heat transfer coefficients for the internal convective elements can be determined using equation (3.2). These heat transfer coefficients are used in the Finite Element Method equations. The equations are solved yielding a temperature distribution. The iteration is repeated until convergence is met, just as in the finned-tapered case.

When convergence is met, that is equation (3.3) has been satisfied, a new film thickness $\delta^*(x)$ can be found by one of the following equations for $\delta^*(x)$:

$$Sh_x \left[\frac{\delta^*}{x} \right]^4 - \frac{1}{3} Dr_x \left[\frac{\delta^*}{x} \right]^3 - \frac{1}{4} Re_{v_x} \left[\frac{\delta^*}{x} \right]^2 - 1 = 0 \quad (\text{eqn 3.7})$$

or

$$Sh_x \left[\frac{\delta^*}{x} \right]^4 - 1 = 0 \quad (\text{eqn 3.8})$$

or

$$\delta^*(x) = \frac{3k_f \mu (T_{sat} - T_w)}{2\rho_f^2 \omega^2 r \sin^2 \theta h_{fg}} \left[1 - \left\{ \frac{r}{(r+x \sin \theta)} \right\}^{8/3} \right]^{1/4} \quad (\text{eqn 3.9})$$

$$\text{where } Sh = \frac{\rho_f^2 (\omega^2 r - g) \sin \theta h_{fg} x^3}{4 \mu k_f (T_{sat} - T_w)}$$

$$Dr = \frac{\rho_f \tau_v h_{fg} x^2 \cos \theta}{\mu k_f (T_{sat} - T_w)}$$

$$Rev = \frac{\rho_f v \cos \theta}{\mu}$$

g = acceleration due to gravity (ft/hr²)

v = vapor velocity (ft/hr)

T_w = local wall temperature (degrees F)

τ_v = shear stress vapor-liquid interface (lbf/ft²)

Equation (3.7) defines the film thickness distribution for a smooth tapered rotating heat pipe derived by Daniels and Al-Jumaily [Ref. 13]. This equation takes into account the drag effects of counter-flowing vapor. Equation (3.8) is a modification of Equation (3.7) neglecting the drag losses. Equation (3.9) was developed by Ballback [Ref. 1]. This equation also neglects drag.

Depending on a particular control parameter which is part of the input data (See Appendix B) one of these equations is used to solve for the film thickness ($\delta^*(x)$) for the next

increment. In addition, the solution temperature distribution from the previous iteration is used as the starting temperature distribution for the next increment.

This iterative process at each increment is repeated until convergence is met, and is continued at each increment until the entire length of the condenser has been transversed, just as in the finned-tapered case. Total heat transfer rates are then determined by summing the incremental heat transfer rates for the entire length of the condenser by the following relationship:

$$Q_{\text{total}} = 360 * \sum_{i=1}^{\text{NDIV}} Q_i \Delta x \quad (\text{eqn 3.10})$$

D. SMOOTH CYLINDRICAL CONDENSER SOLUTION

As in the smooth-tapered case, the condenser is first divided axially, then it is divided circumferentially into 360 segments of equal length. These segments are the basic symmetric unit section to be considered. Again the symmetric unit is subdivided into linear triangular elements and x and y coordinates are assigned to the system nodal points.

To begin the iteration, an initial temperature estimate which is an input parameter is assigned to the convective surface nodal points. Using this initial temperature estimate, the maximum film thickness, δ_{max}^* which is located at $x = 0$, is determined. This maximum film thickness value is then

used as one of the boundary conditions in the solution of equation (2.18) using the Finite Element Method. Equation (2.18) is repeated here for reference.

$$\delta^* \frac{d}{dx} \left[\frac{d}{dx} \delta^{*3} \right] = - \frac{3k_f(T_{sat} - T_w)\mu}{\rho_f^2 \omega^2 r \bar{h}_{fg}} \quad (\text{eqn 3.11})$$

The finite element solution of Equation (3.11) provides the film thickness profile ($\delta^*(x)$) at the midpoint of each increment along the length of the condenser, but only the first increment value is applied at this point of the analysis. Once $\delta^*(x)$ is known at the first increment the heat transfer coefficients for the internal convective surface elements can be determined using equation (3.2). The steady state heat conduction problem is then solved and a temperature distribution for the unit section results. This process is now repeated. A new maximum film thickness, film profile and thus $\delta^*(x)$ at the first increment is found based on the solution temperature distribution from the first iteration. With this new value for $\delta^*(x)$, the heat conduction problem is again solved for a new temperature distribution. This iterative process, involving solution of the film profile with each iteration is continued until temperature convergence is met at the first increment. When convergence is met, the film profile determined on the iteration in which convergence was met is used to provide the values of $\delta^*(x)$ for the remaining increments along the length of the condenser.

Using this predetermined value of $\delta^*(x)$, the iterative process is continued at each increment until temperature convergence is reached. As convergence is reached at each increment, the internal wall temperature, which is the same for all internal wall nodal points of the section, is stored for future application.

When convergence is reached at the final increment, equation (3.11) is once again solved for the film profile. It should be noted, however, that the right hand side of equation (3.11) is temperature dependent. It should also be noted that the temperature varies axially along the length of the condenser. In order to account for this temperature dependence and temperature variation, the right side of equation (3.11) is now determined for each increment using the wall temperatures that were stored at each increment. The finite element solution of the film profile will now account for the temperature variation along the length of the condenser.

This final film profile provides the value of $\delta^*(x)$ for each increment. The iterative process of solving for heat transfer coefficient, temperature distribution, and temperature convergence is continued for each increment until the final increment is reached.

Equation (2.13) provides a relationship for total mass flow rate as a function of position. The total mass flow rate at the overfall into the evaporator is given by:

$$\dot{m} = - \frac{2\pi\rho_f^2\omega^2r}{\mu} \cdot \frac{d\delta^*(L)}{dx} \cdot \frac{\delta^{*3}(L)}{3} \quad (\text{eqn 3.12})$$

where $\delta^*(L)$ and $d\delta^*(L)/dx$ are the values for the film thickness and rate of change of film thickness with respect to x at the evaporator end of the condenser. Another relationship for mass flow rate based on the steady state heat conduction solution is given by:

$$\dot{m} = 360 \cdot \sum_{i=1}^{NDIV} \frac{Q_i \Delta x}{\bar{h}_{fg}} \quad (\text{eqn 3.13})$$

If, in fact, the solution of equations (3.12) and (3.13) are equal, then the mass flow rate of the condensate returning to the evaporator is equal to the mass flow rate of the vapor being condensed on the surface of the condenser. Or, to put it another way, continuity is satisfied.

It should be noted that the film profile maintains the same basic shape, that is, $\delta^*(x)$ at $x=0$ is always equal to δ_{\max}^* and decreases to a specific minimum value at $x=L$. This being the case, if the maximum film thickness is varied, the film thickness profile will vary in the same manner. For example, if the maximum film thickness is increased, the entire profile will also increase. This will result in an increased internal thermal resistance and thus lower heat transfer rate. As a result of the lower heat transfer rate, the mass flow rate as

determined by equation (3.13) will be less. At the same time, however, the greater film profile will result in a greater value of $\delta^*(x)$ at the overfall, $x = L$. Yet, since the profile maintained the same basic shape, the derivative at the overfall remains relatively constant. Thus the mass flow rate as determined by equation (3.12) will increase. A decrease in the maximum film thickness will result in an opposite effect to the mass flow rates determined by equations (3.12) and (3.13).

This being the case, the mass flow rates, as determined by equations (3.12) and (3.13) are now compared to determine if the film profile is in fact the solution profile to the problem. If continuity is not satisfied, the maximum film thickness is varied and the entire iterative process is restarted. This process is continued until the film profile mass flow rate, equation (3.12) converges towards the heat transfer mass flow rate, equation (3.13). When the absolute difference between these mass flow rates is less than a specific value, the resulting heat transfer rate is considered the solution to the problem.

E. FINNED CYLINDRICAL CONDENSED SOLUTION

As in the finned-tapered case, the condenser is divided axially and circumferentially into the basic symmetric section as shown in Figure 1. This unit is then subdivided into linear triangular elements and x and y coordinates are assigned to each nodal point.

To begin the iterative process, as before, an initial temperature estimate is assigned to each nodal point along the internal convective surface. An initial trough film profile is determined, using equation (3.11) and this initial temperature estimate. In this case, however, the maximum film thickness (δ_{\max}^*) is not calculated but is an input parameter.

Once $\delta^*(x)$ is known at the first increment, the internal heat transfer coefficients are determined, using equations (3.1) and (3.2). These values are then used in the Finite Element Method solution of the steady state heat conduction problem. A temperature distribution is determined and the iteration is repeated until temperature convergence is met. Note that a new film profile is determined for each iteration.

As in the cylindrical-smooth condenser case, once convergence is met at the first increment, the film profile that was determined for the iteration prior to convergence at the first increment is then used to provide the film thickness $\delta^*(x)$ for the remaining increments.

At the final increment, a new film profile for a finned cylindrical condenser is then determined by solving the following equation developed in Chapter II:

$$\delta^* \frac{d}{dx} \left[\frac{d\delta^*}{dx} (\delta^{*3} \epsilon + \delta^{*4} \tan \alpha) \right] = - \frac{3k_f(T_{\text{sat}} - T_w) \mu \epsilon}{\rho_f^2 \omega^2 r \bar{h}_{fg}}$$

$$- 2\delta^* \cos \alpha \left[\frac{4k_f(T_{\text{sat}} - T_{\text{avg}}) \mu z^*}{\rho_f^2 \omega^2 r h_{fg} \cos \alpha} \right]^{3/4} \quad (\text{eqn 3.14})$$

A description of the solution of this differential equation using the Finite Element Method is provided in Appenxix A.

The iterative process of finding the solution temperature distribution for all increments is then repeated until the length of the condenser has been transversed.

At this point, the total mass flow rate of the condenser returning to the evaporator is determined by the following relationship:

$$\dot{m} = -\frac{\rho_f^2 \omega^2 r}{3\mu} \frac{d\delta^*}{dx} (\epsilon \delta^{*3} + \delta^{*4} \tan \alpha) * ZFIN \quad (\text{eqn 3.15})$$

where ZFIN is the number of axial fins.

This mass flow rate is compared to the mass flow rate given by the following equation:

$$\dot{m} = ZFIN * 2 * \sum_{i=1}^{NDIV} \frac{Q_i \Delta x}{h_{fg}} \quad (\text{eqn 3.16})$$

Just as in the smooth-cylindrical condenser case, if the absolute difference between the two mass flow rates is less than a mass flow convergence criterion, the problem is considered solved. If not, δ_{\max}^* is varied and the entire iterative process is started again. As in the smooth-cylindrical condenser, varying δ_{\max}^* will have the same effect on the film profile and the heat transfer rate. Since the temperature distribution along the condenser has been solved once and closely

approximates the final solution to the problem, on successive iterations, equation (3.14) may be used to solve for the film profile rather than equation (3.11). Equation (3.11) was used on the first iteration because the finite element solution converges more quickly than equation (3.14) when an estimated temperature is used.

A word of caution is required. The solution of equation (3.14) is highly sensitive to the value of δ_{\max}^* . If the initial value of δ_{\max}^* is inconsistent with the actual solution, e.g. too small, the Finite Element Method solution of the film profile will not converge. If this is in fact the case, the problem will be automatically terminated. A new value of δ_{\max}^* should then be chosen and the problem restarted.

One additional topic which should be addressed is the rectangular fin solution process. The rectangular fin profile is a slight modification of the finned-tapered or finned-cylindrical condenser solution. The only variation is that the top elements of the rectangular fin are assigned heat transfer coefficients of 0.0. Thus, the tip of the rectangular fin is considered adiabatic and no heat is transferred through this top face. Other than this modification, the solution techniques are the same as for the finned cases addressed above.

IV. RESULTS AND DISCUSSION

Prior to the heat transfer analysis of the various condenser configurations, it was necessary to verify the finite element solution of the film profile. The development of this solution is discussed in Appendix A.

Leppert and Nimmo [Refs. 8 and 9] had developed an analytical solution for film condensation on a horizontal plate at a constant surface temperature. Their analysis and resulting differential equation is identical to the development in Chapter II for a smooth cylindrical condenser if the acceleration due to gravity is replaced by a radial acceleration term. This being the case, this modification was made and the analytical solution was used as a reference for comparison.

For the test runs of the finite element solution, a constant surface temperature was assumed. In addition, a value for the maximum film thickness (δ_{\max}^*) and minimum film thickness (δ_{\min}^*) at the overfall were required. The value for δ_{\max}^* was determined based on a relationship developed by Leppert and Nimmo [Ref. 8]. The value for δ_{\min}^* was arbitrarily chosen.

For identical geometry, surface temperature and maximum and minimum film thicknesses, the results of both analyses were identical. In order to develop confidence in the finite element solution, δ_{\min}^* was varied from $0.10 \cdot \delta_{\max}^*$ to $0.97 \cdot \delta_{\max}^*$. The resulting profiles agreed at all locations along the length of

the condenser. However, when the temperature was permitted to vary along the length of the condenser in the finite element solution and a corresponding average temperature was used in the analytical solution of Leppert and Nimmo [Refs. 8 and 9], the profiles were no longer in agreement. This was to be expected, particularly in the case where there was a sixteen degree Fahrenheit variation along the length of the condenser. This substantial temperature variation resulted in a significant variation of fluid properties which would account for a difference in film profile.

Due to the agreement between the finite element solution and the analytical solution of Leppert and Nimmo [Refs. 8 and 9] for a constant surface temperature, it was decided that the finite element solution does provide a satisfactory representation of the film profile. This being the case, the finite element solution was then incorporated into the code to provide the film profile for the cylindrical condenser.

Once the finite element solution of the film profile was verified, the heat transfer analysis could be accomplished. The analysis considered both copper and stainless steel condensers with the following four geometries: a) tapered-smooth, b) tapered-axially finned, c) cylindrical-smooth and d) cylindrical-axially finned. Table I lists the geometric parameters held constant for all analyses. In all cases, the working fluid was water.

TABLE I

Condenser Geometric Parameters
Held Constant During All Analyses

condenser length	=	8.500	inches
minimum radius	=	0.51575	inches
wall thickness	=	0.05118	inches

In addition, the following geometric parameters were also utilized when required. This requirement was based on the condenser geometry being considered, i.e., tapered-axially finned.

TABLE II

Condenser Geometric Parameters
Applied as Required

height of fin	=	0.05906	inches
fin half angle	=	26.565	degrees
condenser cone half angle	=	1.00	degrees

In the analysis, the heat transfer rate was determined for the four different geometries listed above for both copper and stainless steel. The ambient temperature was set at 60.0°F and the heat transfer rate was determined for each possible combination of the operating parameters given in Table III.

TABLE III
Operating Parameter Matrix

Rotational Speed (RPM)	Heat Transfer Coefficient (btu/hr-ft ² -F)	Saturation Temperature (degree F)
700.0	100.0	90.0
1400.0	500.0	120.0
2800.0	1000.0	150.0
		180.0

Thus, for each condenser geometry, there was a total of 72 analyses, 36 for the case of the condenser with a copper wall and 36 for the case of the condenser with a stainless steel wall.

The first condenser geometry considered was a smooth condenser. Figures 8-13 compare the heat transfer rates of smooth cylindrical condensers with those of smooth tapered condenser. In particular, Figures 8,9, and 10 indicate the results of the analyses of smooth copper condensers at rotational speeds of 700, 1400 and 2800 revolutions per minute(RPM) respectively. Figures 11, 12, and 13 are for smooth stainless steel condensers at 700, 1400, and 2800 RPM respectively. For both stainless steel and copper smooth condensers, the following general observations apply: a) For the same external heat transfer coefficient, the heat transfer rate for the cylindrical smooth condenser is less than the equivalent tapered condenser. b) As the external heat transfer coefficient increases, this difference

in heat transfer rate also increases. c) The rotational speed has a greater effect on the tapered condenser heat transfer rate. For example, the maximum heat transfer rate of a copper tapered condenser will increase by a factor of 1.67 when the rotational speed is increased from 700 to 2800 RPM. In the cylindrical copper condenser, for the same change in rotational speed, the heat transfer rate only increases by a factor of 1.51.

These same observations hold true for the smooth stainless steel condensers. But, due to a greater thermal resistance in the wall of the stainless steel condenser, the heat transfer rates are less for all cases considered. It should be noted that the thermal conductivity of stainless steel is only 4% the thermal conductivity of copper. This accounts for the increased thermal resistance.

Figures 14, 15, and 16 compare axially finned cylindrical with axially finned tapered copper condensers at 700, 1400 and 2800 RPM respectively. Note that the heat transfer rates of the cylindrical condensers are only slightly less than those of the tapered condensers. This is because the heat is primarily transferred through the extended surface, i.e., the fin. Thus the film condensate in the trough has less effect on the heat transfer rate. In fact, the average difference in heat transfer rate for 700 RPM and an external heat transfer coefficient of 100 Btu/hr-ft²-F is 12.65%. This difference increases to 15.6% as the heat transfer coefficient is increased to 1000 Btu/hr-ft²-F. However, as the rotational speed is increased to 2800

RPM, the corresponding average differences in heat transfer rates decrease to 12.58% and 13.33% respectively.

Figures 17, 18, and 19 compare the heat transfer rates of axially finned stainless steel cylindrical condensers with those of axially finned stainless steel tapered condensers at 700, 1400 and 2800 RPM respectively. Note the difference in heat transfer rate increases as the external heat transfer coefficient increases more than in the case of the copper condensers. At the low heat transfer coefficient, the limiting thermal resistance is that of the external surface ($1/h_{ext}$). However, as the external heat transfer coefficient is increased, the limiting thermal resistance becomes that of the wall due to the low thermal conductivity of stainless steel.

Another observation to be noted is the fact that the rotational speed has a greater effect on the heat transfer rate of the cylindrical condensers than on the tapered condensers. As the rotational speed increases, the film thickness decreases due to the greater centrifugal force exerted on the film. For the cylindrical condenser, the maximum film thickness is much greater than for a tapered condenser. For example, an axially finned stainless steel cylindrical condenser rotating at 1400 RPM with an external heat transfer coefficient of 1000 Btu/hr-ft²-F has a maximum film thickness twice that of a corresponding tapered condenser. This being the case, higher rotational speeds will have a greater effect on the greater film thickness and the difference in heat transfer rates will decrease.

Figures 20, 21, and 22 compare the heat transfer rates of copper smooth cylindrical condensers with copper axially finned cylindrical condensers at 700, 1400 and 2800 RPM respectively. The figures indicate that for low external heat transfer coefficients, little is to be gained by the addition of axial fins. But, as the external heat transfer coefficient increases, the advantage becomes significant. As an example, consider Figure 21. For $h=100 \text{ Btu/hr-ft}^2\text{-F}$, axial finning increase the heat transfer rate by 16%. But, for an external heat transfer coefficient of $1000 \text{ Btu/hr-ft}^2\text{-F}$, the heat transfer rate increases by 194%.

Note also, that as rotational speed increases, the advantage to be gained by axial finning decreases. This can be explained by the fact that for an axially finned condenser, the majority of the heat is transferred by the fin surface. As the rotational speed increases, the film thickness in the trough decreases. This in turn will expose slightly more fin surface area. On the other hand, for the smooth condenser, the decrease in the film thickness will have a greater effect on heat transfer rate in that the thermal resistance of the film has decreased. Comparing the two geometries, the change in overall thermal resistances for the smooth condenser will be greater than that for the axially finned condenser accounting for the slight decrease in advantage with increasing rotational speed.

Figures 23, 24, and 25 correspond to Figures 20, 21, and 22 but for stainless steel condensers. The results are similar to the copper situation discussed above, by the effect caused by the increasing external heat transfer coefficient is not as dominant due to the high thermal resistance of the stainless steel wall material.

Figure 26 compares a smooth cylindrical copper condenser with a smooth cylindrical stainless steel condenser at 1400 RPM. As to be expected, the heat transfer rate of the copper condenser is greater than that of the stainless steel condenser due to the difference in the thermal conductivity of the two materials. The difference in the heat transfer rate is least for a low external heat transfer coefficient where the external thermal resistance is dominant. As the heat transfer coefficient increases, the thermal resistance of the wall of the condenser becomes more important resulting in an increasing difference between the two condensers.

Figure 27 provides a comparison of axially finned cylindrical copper and stainless steel condensers at 1400 RPM. The advantage of copper over stainless steel is obvious. Note that the heat transfer rate for the copper condenser at 500 Btu/hr-ft²-F is nearly identical to the stainless steel condenser heat transfer rate at 1000 Btu/hr/-ft²-F indicating the advantage of copper over stainless steel.

The final analysis compared the heat transfer rate of an axially finned copper condenser with a triangular fin profile

to the heat transfer rate of an axially finned condenser with a rectangular fin profile. The rectangular fin was assumed to have an adiabatic tip. This comparison was accomplished for both cylindrical and tapered condensers. Table IV lists the parameters used in the analysis. These parameters are in addition to those listed in Table I. This comparison was only accomplished for the one set of operating parameters listed in Table IV. Note that the operating parameters chosen were the median values.

TABLE IV

List of Parameters Used in
Rectangular/Triangular Fin Profile Comparison

Heat Transfer Coefficient	=	500 Btu/hr-ft ² -F
Rotational Speed	=	1400 RPM
Saturation Temperature	=	120.0 degrees F
Fin Height	=	0.05906 inches

Table V lists the results of the analyses.

TABLE V

Results of Rectangular/Triangular
Fin Profile Comparison

CONDENSER GEOMETRY	FIN PROFILE	Q (Btu/hr)
tapered	triangle	6168.93
	rectangle	6143.03
cylindrical	triangle	5358.4
	rectangle	5338.00

Note that the heat transfer rate varied by only 0.4% for both the tapered and cylindrical condensers. The reason for this insignificant variation is provided in Table VI and VII.

Table VI lists the convective surface temperature distribution of a symmetric unit section of a tapered condenser at the middle increment of the condenser. Temperature location 1 is located at the tip of the fin. For the rectangular profile, temperature location 1 is located at the intersection of the adiabatic surface (the tip of the fin), and the vertical surface of the fin. Temperature location 5 is located at the base of the fin. Locations 6-9 are located in the trough region and the remaining temperature locations are along the external surface of the section. Thus temperatures 1-5 provide the temperature distribution along the convective surface of the fin. Note that the temperature distribution is lower for the rectangular fin.

Thus, the driving force for heat transfer, the temperature difference between the saturation temperature and the surface temperature is greater for the rectangular fin. Thus, in spite of the fact that the fin surface area for heat transfer has decreased by 11%, the average surface temperature difference has increased by 25%. It should also be noted that the average heat transfer coefficient for the fin surface for the rectangular fin also decreases by 12%. However, the increase in temperature difference is the dominant change that only allows a 0.4% decrease in heat transfer rate.

TABLE VI

Surface Temperature Distribution for Rectangular
and Triangular Axially Finned Tapered Copper
Condensers at Middle Increment of Condensers

Temperature Location	Triangular Fin Temperature (F)	Rectangular Fin Temperature (F)
1	118.85	117.95
2	118.45	117.88
3	118.04	117.68
4	117.60	117.36
5	117.05	116.59
6	116.82	116.45
7	116.61	116.35
8	116.59	116.35
9	116.48	116.54
10	116.47	116.53
11	116.46	116.23
12	116.46	116.21
13	116.44	116.20
14	116.42	116.17
15	116.31	116.12
16	116.31	116.07
17	116.27	116.04
18	116.26	116.03

TABLE VII

Surface Temperature Distribution for Rectangular
and Triangular Axially Finned Cylindrical Copper
Condensers at Middle Increment of Condensers

Temperature Location	Triangular Fin Temperature (F)	Rectangular Fin Temperature (F)
1	118.55	117.55
2	118.09	117.44
3	117.62	117.25
4	117.10	116.87
5	116.55	116.36
6	116.38	116.18
7	116.28	116.07
8	116.22	116.01
9	116.20	115.99
10	116.03	115.80
11	116.02	115.79
12	116.01	115.79
13	115.99	115.77
14	115.97	115.75
15	115.94	115.72
16	115.91	115.67
17	115.88	115.67
18	115.88	115.66

Table VII indicates a similar situation exists in the cylindrical condenser. Again note the decrease in the average surface temperature of the fin for the rectangular profile. The surface area and heat transfer coefficient has decreased, but the increased temperature difference compensates for these changes, limiting the overall decrease in heat transfer rate.

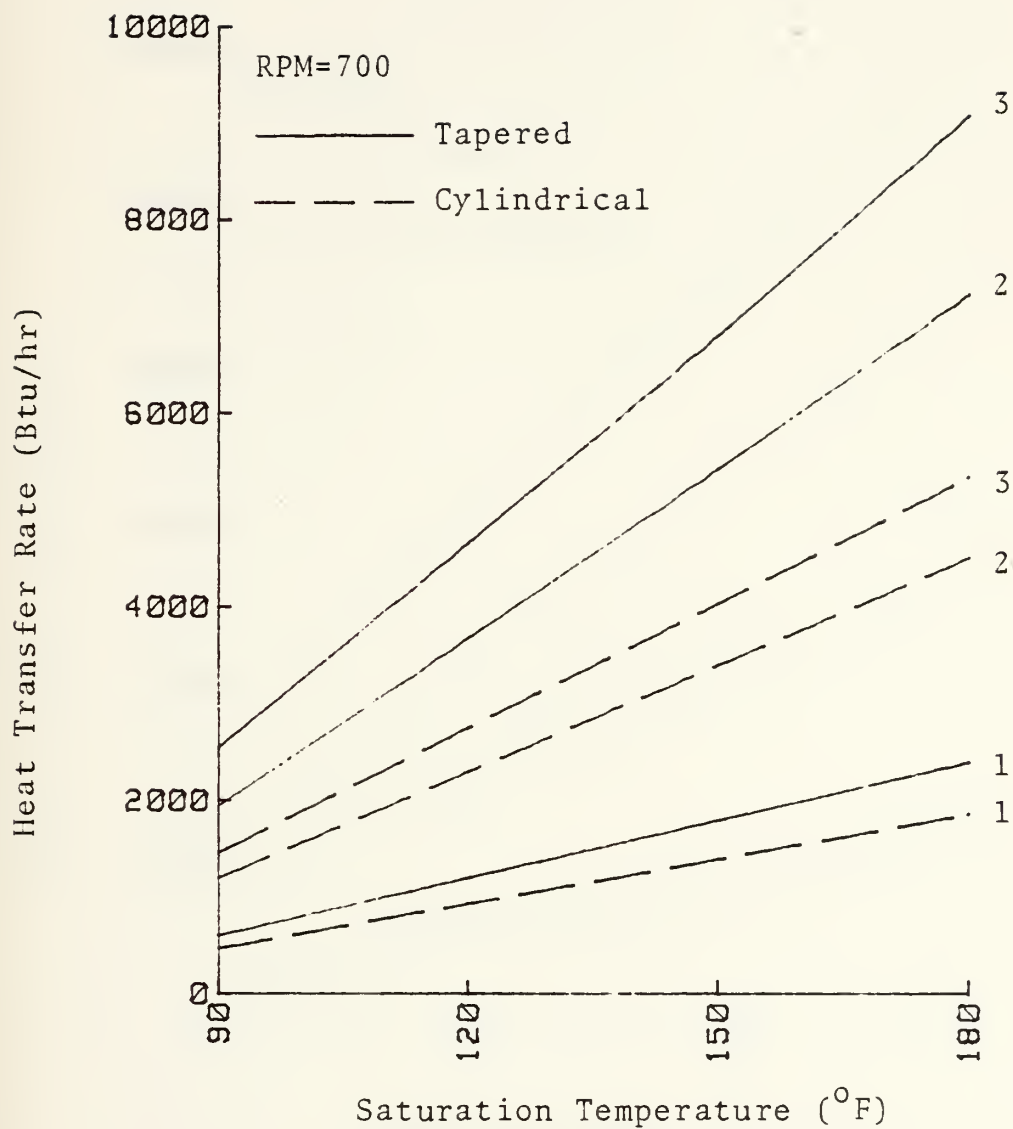
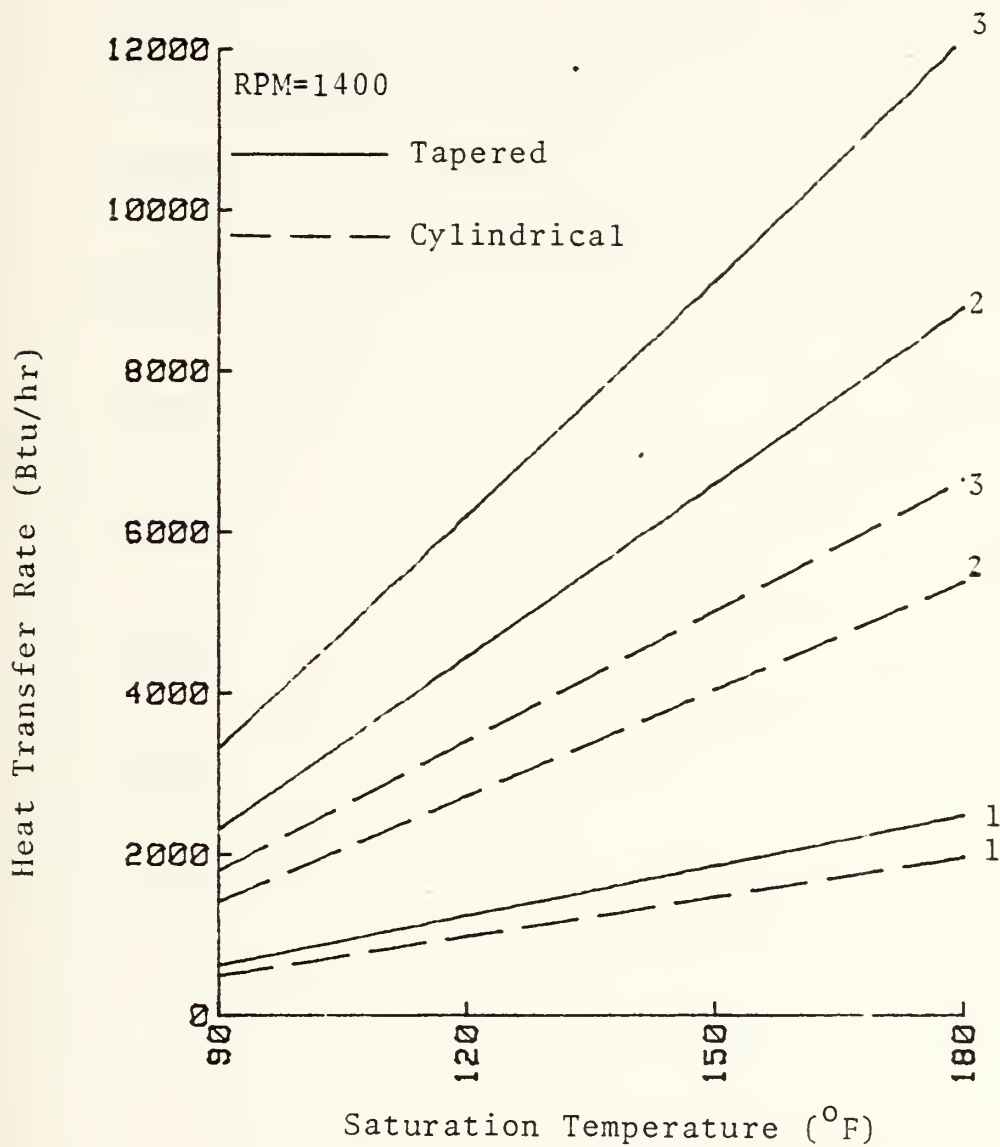
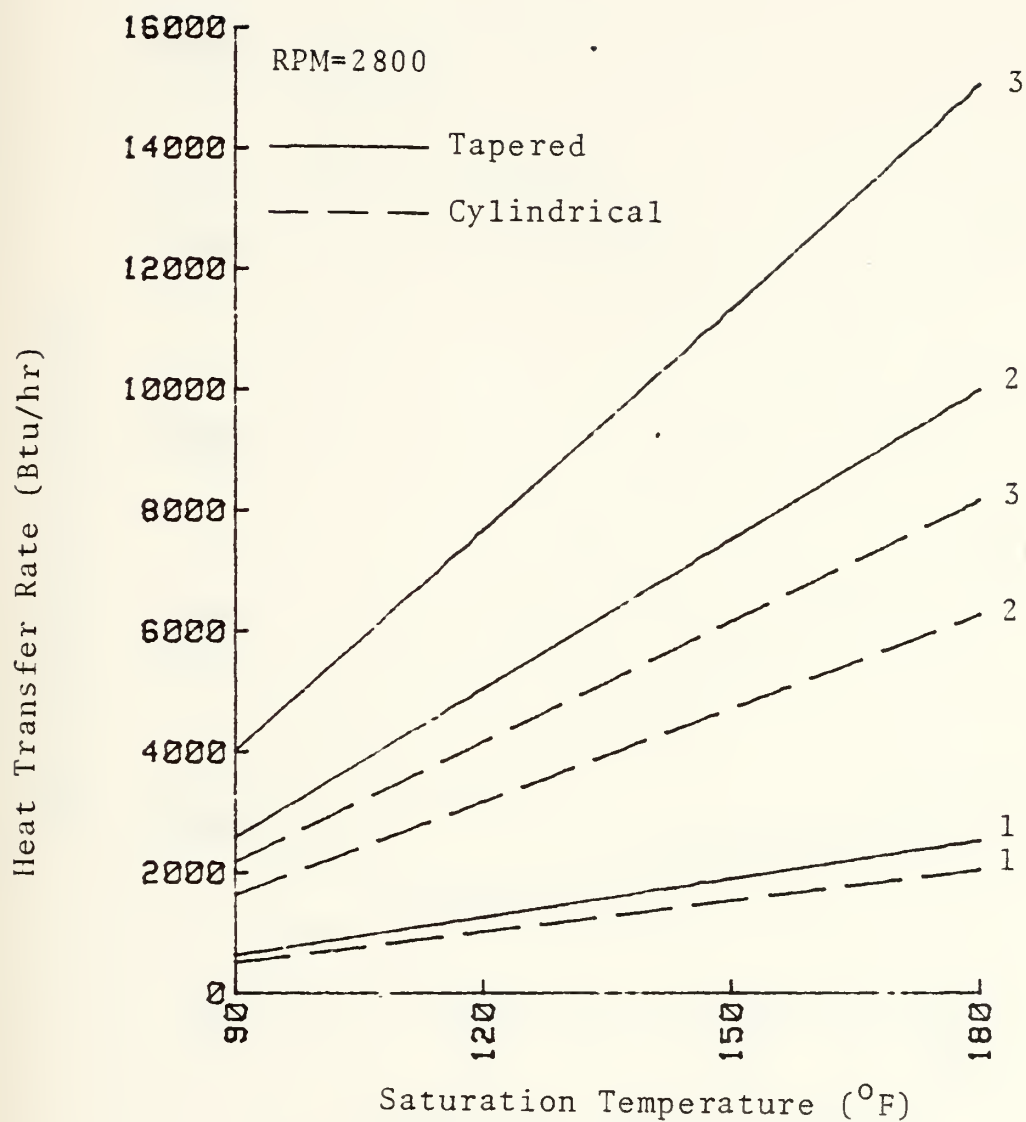


Figure 8. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Copper Condensers at 700 RPM



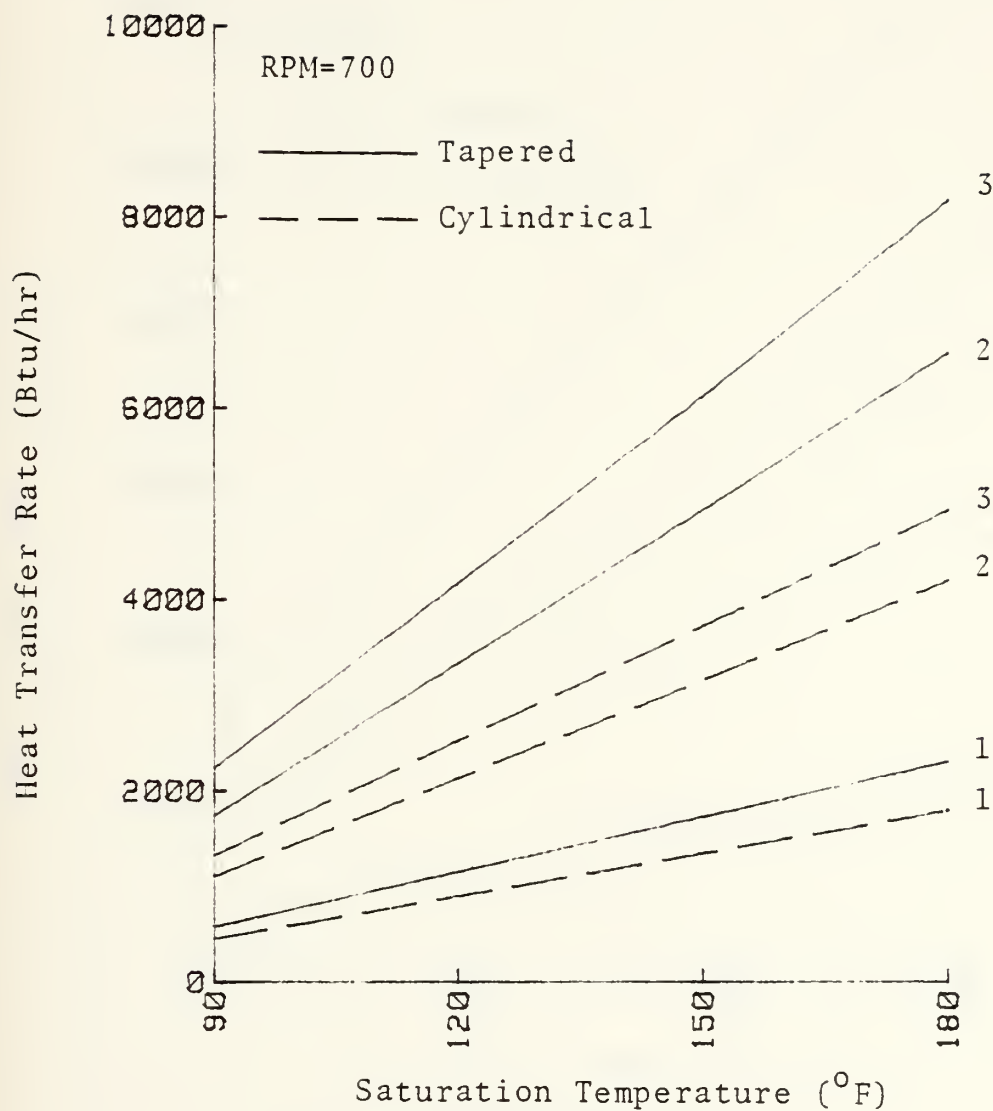
1. $h_{ext} = 100 \text{ Btu/hr-ft}^2\text{-}^{\circ}\text{F}$
2. $h_{ext} = 500 \text{ Btu/hr-ft}^2\text{-}^{\circ}\text{F}$
3. $h_{ext} = 1000 \text{ Btu/hr-ft}^2\text{-}^{\circ}\text{F}$

Figure 9. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Copper Condensers at 1400 RPM



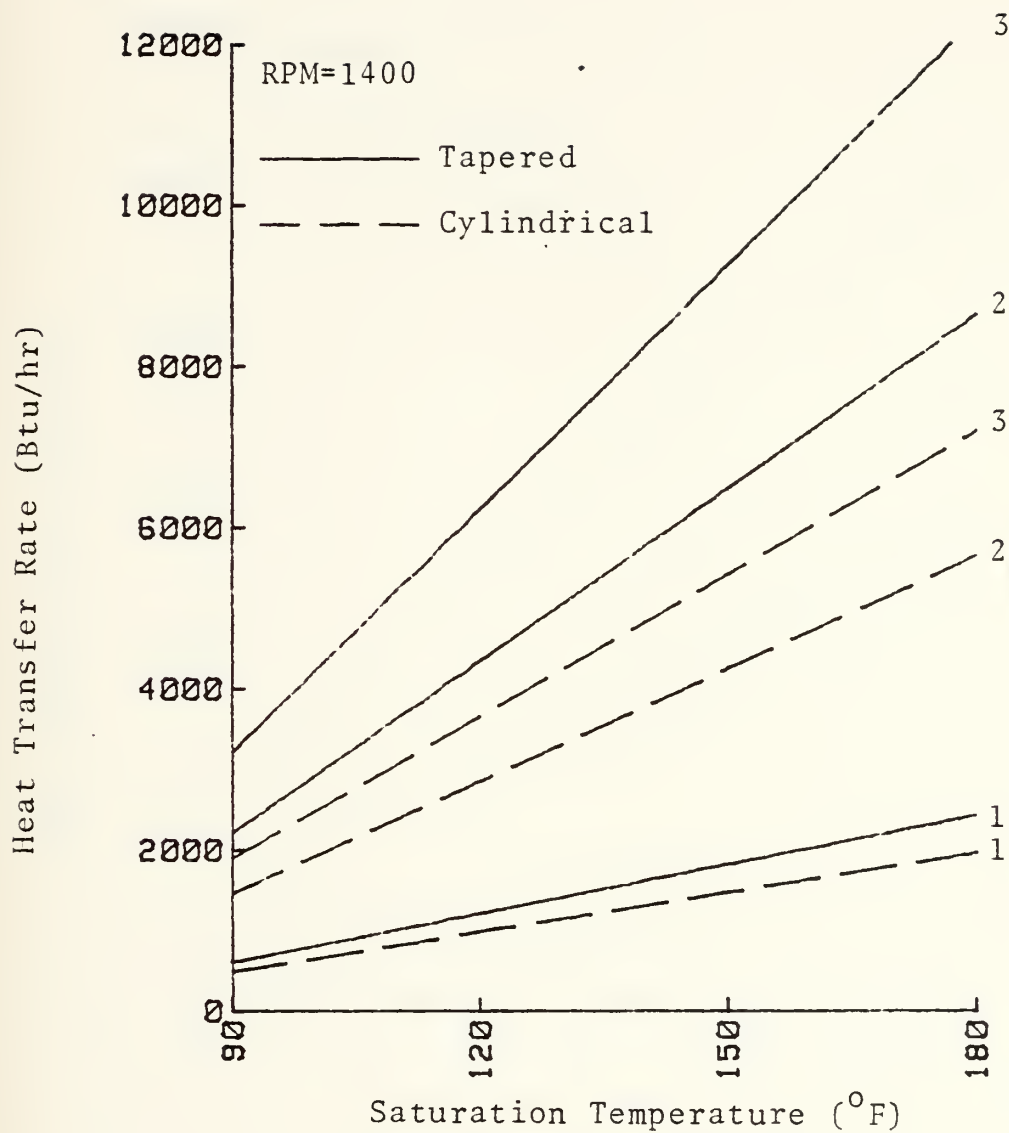
1. $h_{ext} = 100$ Btu/hr-ft²-°F
2. $h_{ext} = 500$ Btu/hr-ft²-°F
3. $h_{ext} = 1000$ Btu/hr-ft²-°F

Figure 10. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Copper Condensers at 2800 RPM



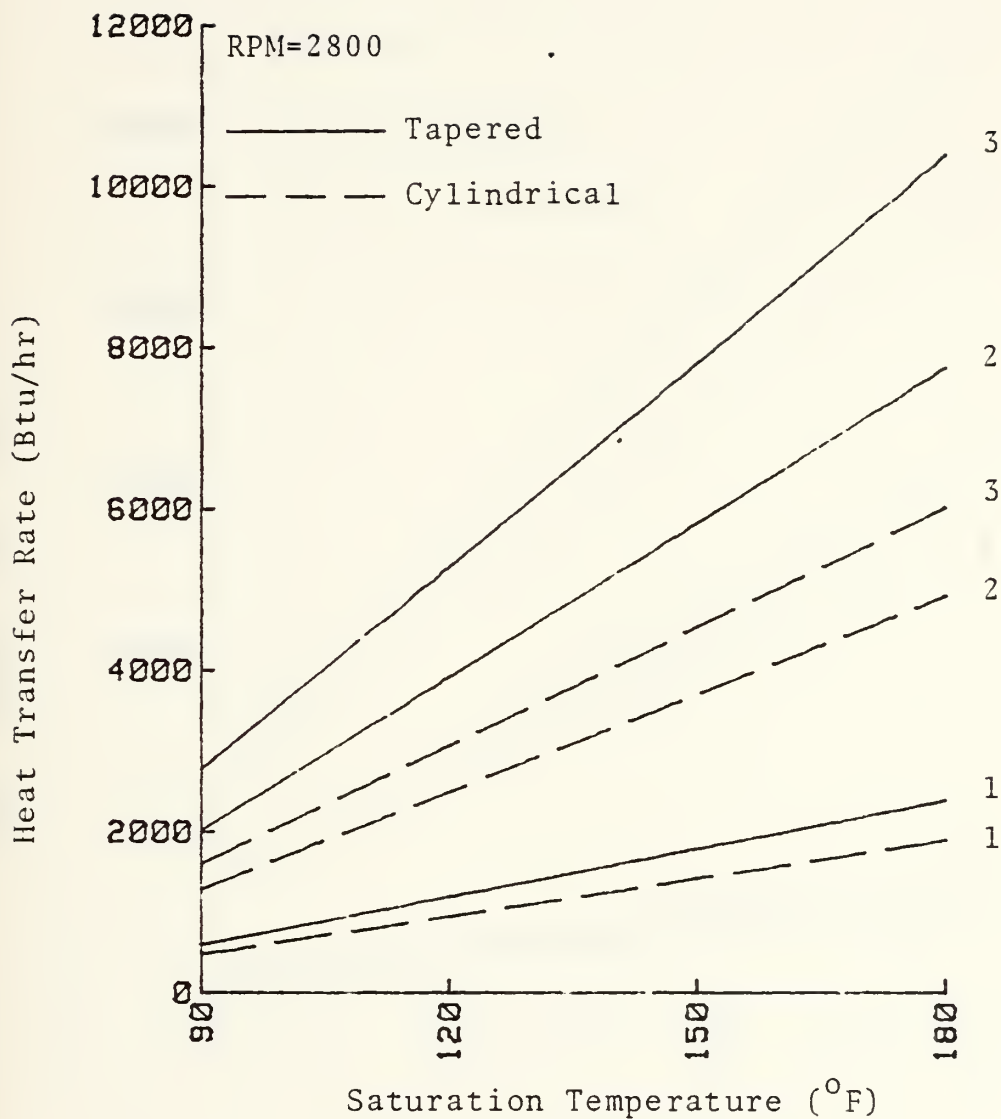
1. $h_{ext} = 100 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
2. $h_{ext} = 500 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
3. $h_{ext} = 1000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$

Figure 11. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Stainless Steel Condensers at 700 RPM



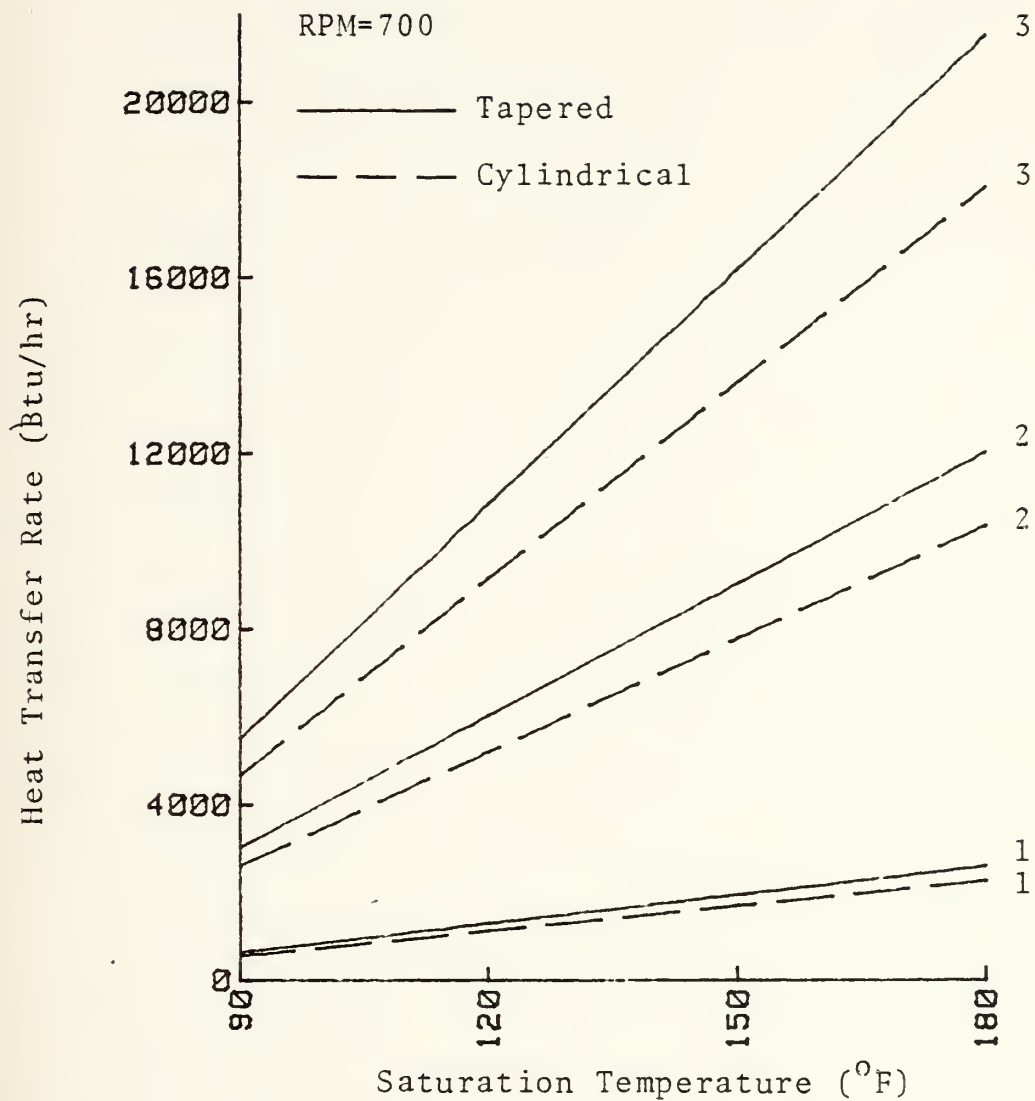
1. $h_{ext} = 100 \text{ Btu/hr-ft}^2\text{-F}$
2. $h_{ext} = 500 \text{ Btu/hr-ft}^2\text{-F}$
3. $h_{ext} = 1000 \text{ Btu/hr-ft}^2\text{-F}$

Figure 12. Heat Transfer Rate versus Saturation Temperature for Smooth Cylindrical and Tapered Stainless Steel Condensers at 1400 RPM



1. $h_{ext} = 100 \text{ Btu/hr-ft}^2\text{-F}$
2. $h_{ext} = 500 \text{ Btu/hr-ft}^2\text{-F}$
3. $h_{ext} = 1000 \text{ Btu/hr-ft}^2\text{-F}$

Figure 13. Heat Transfer Rate versus Saturation Temperature for Smooth Tapered and Cylindrical Stainless Steel Condensers at 2800 RPM



1. $h_{ext} = 100 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
2. $h_{ext} = 500 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
3. $h_{ext} = 1000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$

Figure 14. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Copper Condensers at 700 RPM

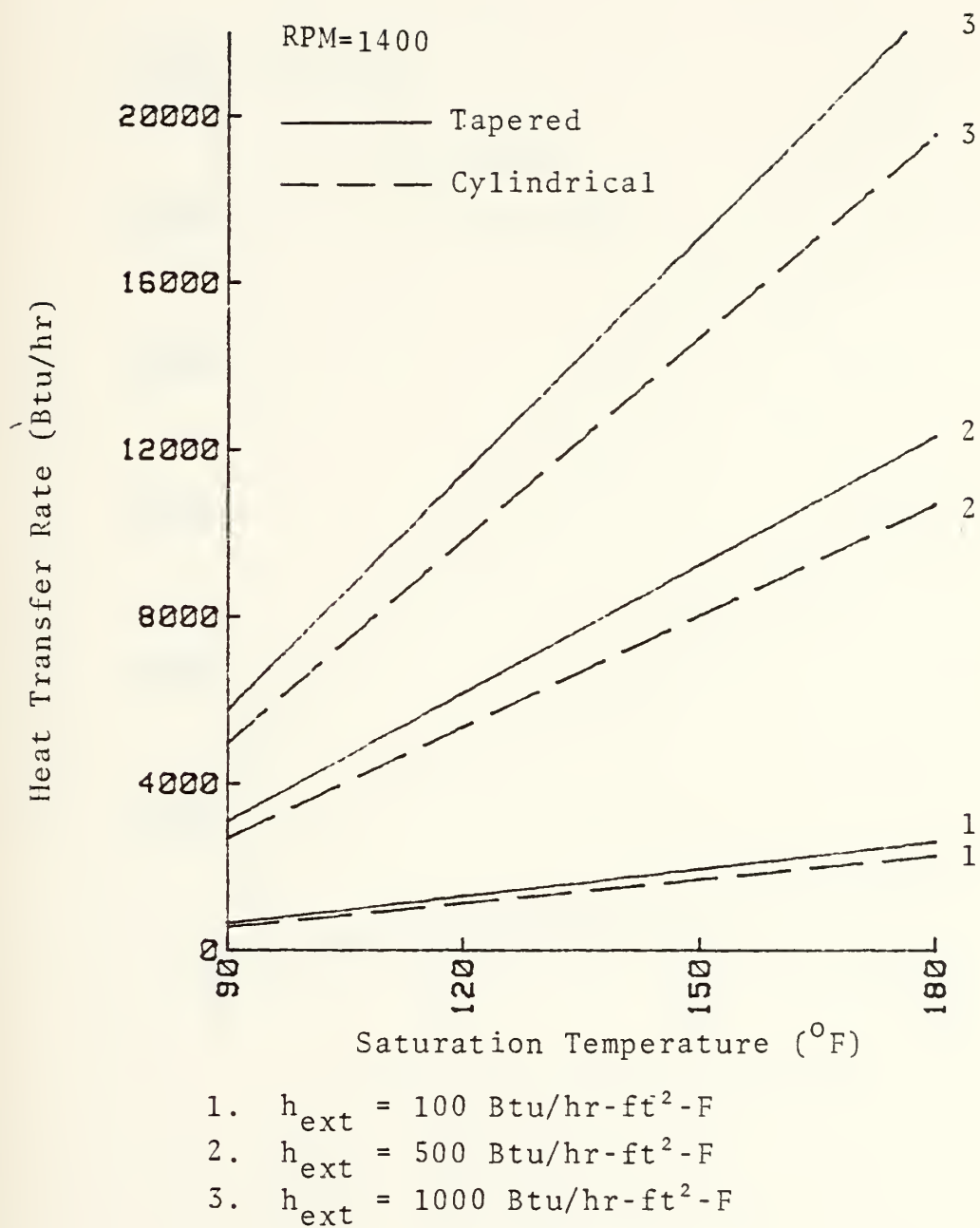
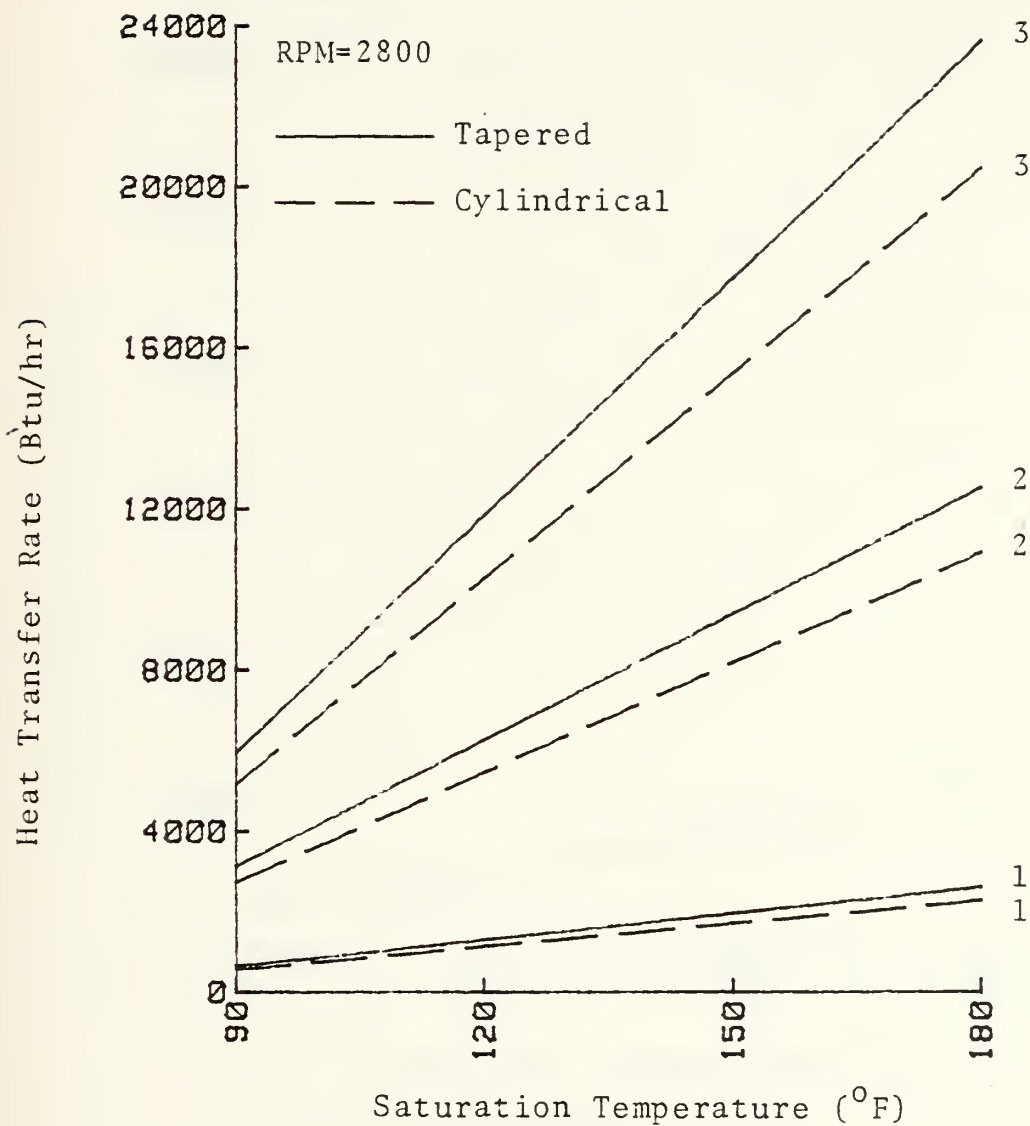


Figure 15. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Copper Condensers at 1400 RPM



1. $h_{ext} = 100 \text{ Btu/hr-ft}^2\text{-}^{\circ}\text{F}$
2. $h_{ext} = 500 \text{ Btu/hr-ft}^2\text{-}^{\circ}\text{F}$
3. $h_{ext} = 1000 \text{ Btu/hr-ft}^2\text{-}^{\circ}\text{F}$

Figure 16. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Copper Condensers at 2800 RPM

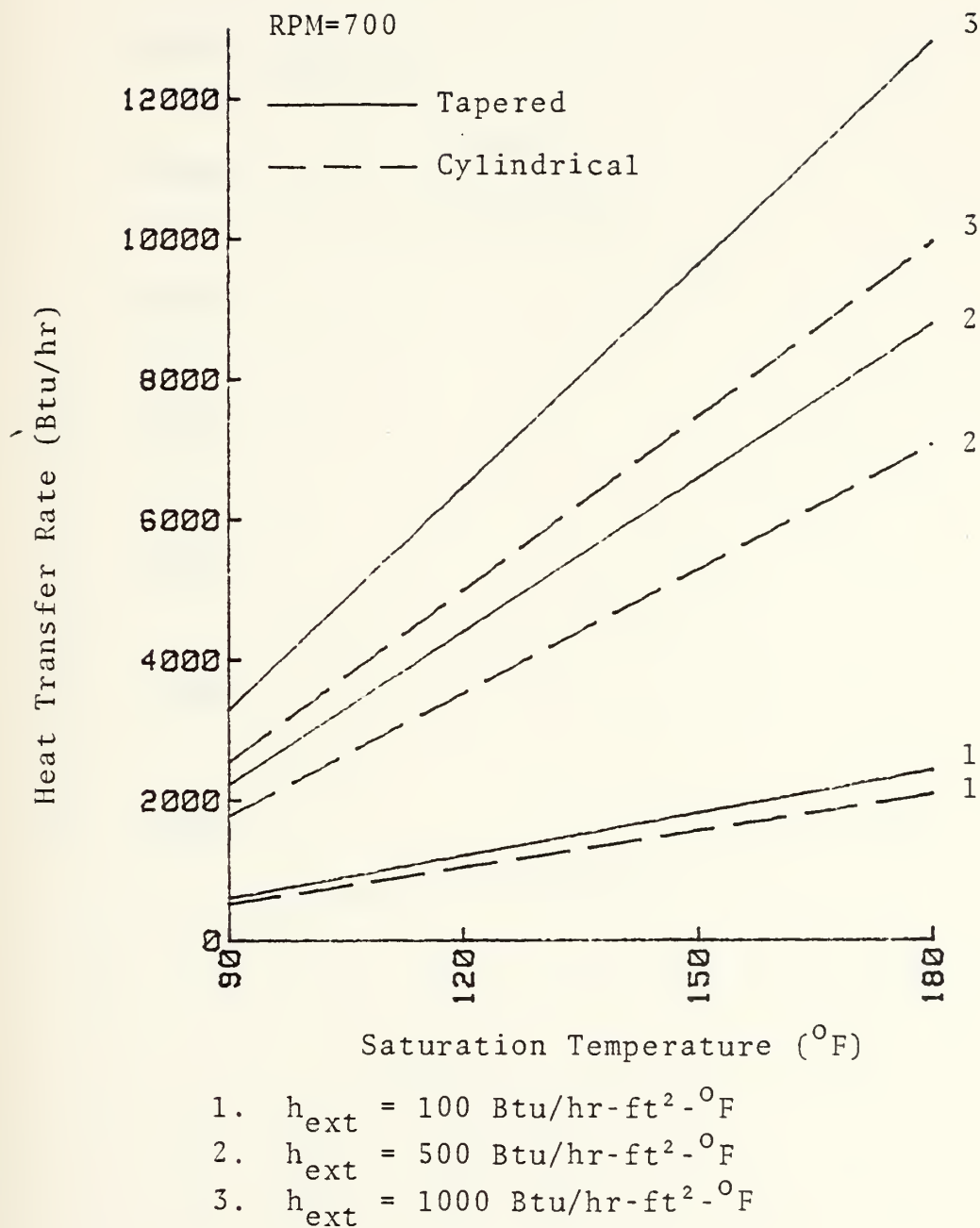
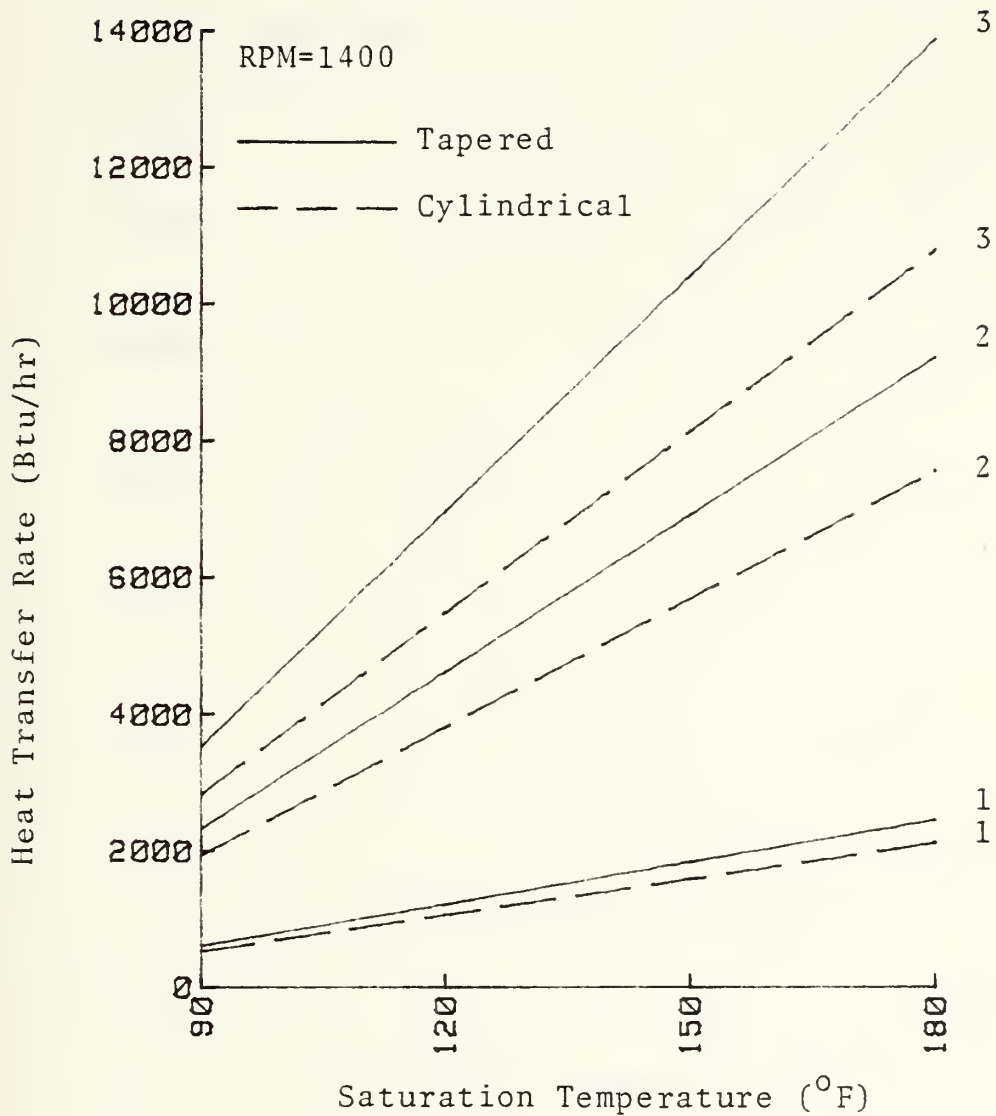
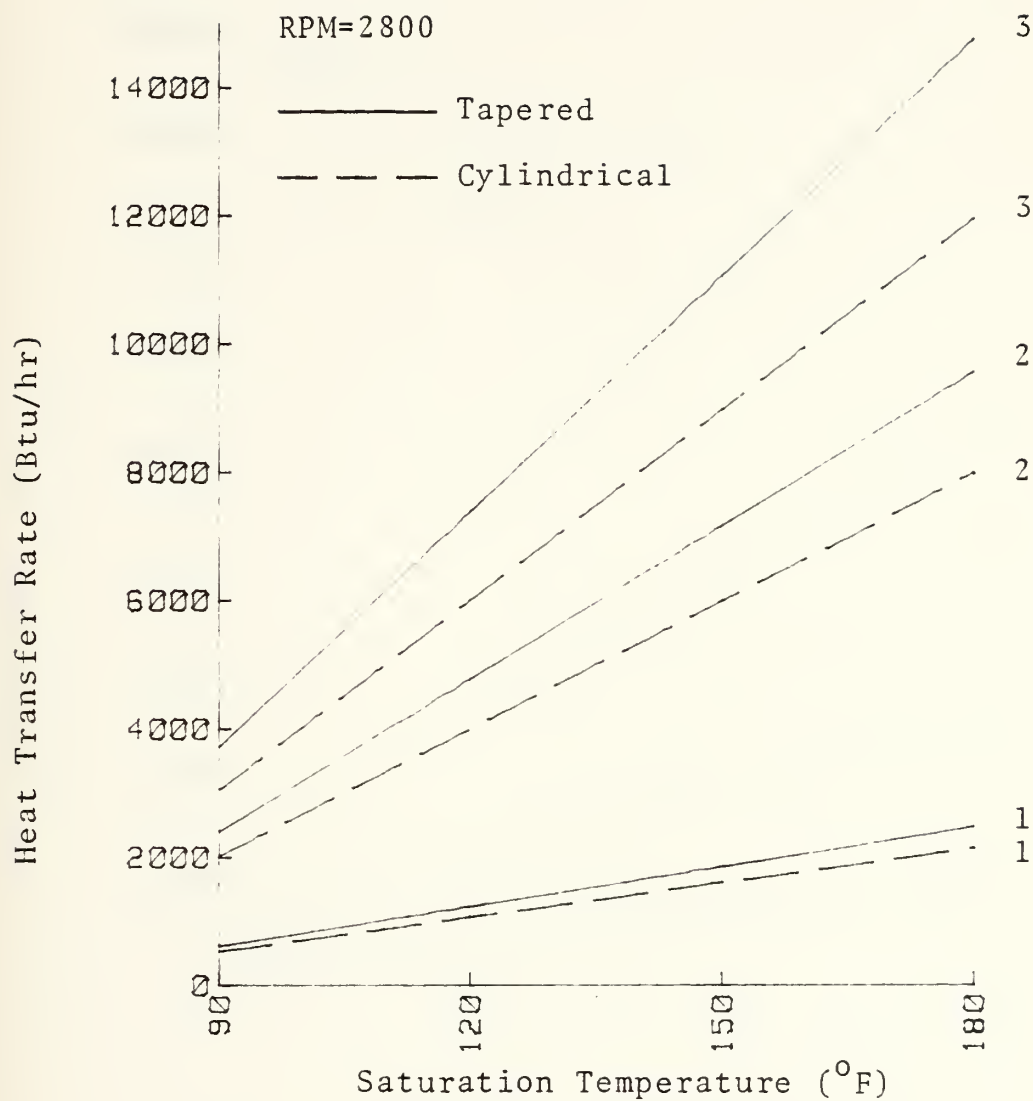


Figure 17. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Stainless Steel Condensers at 700 RPM



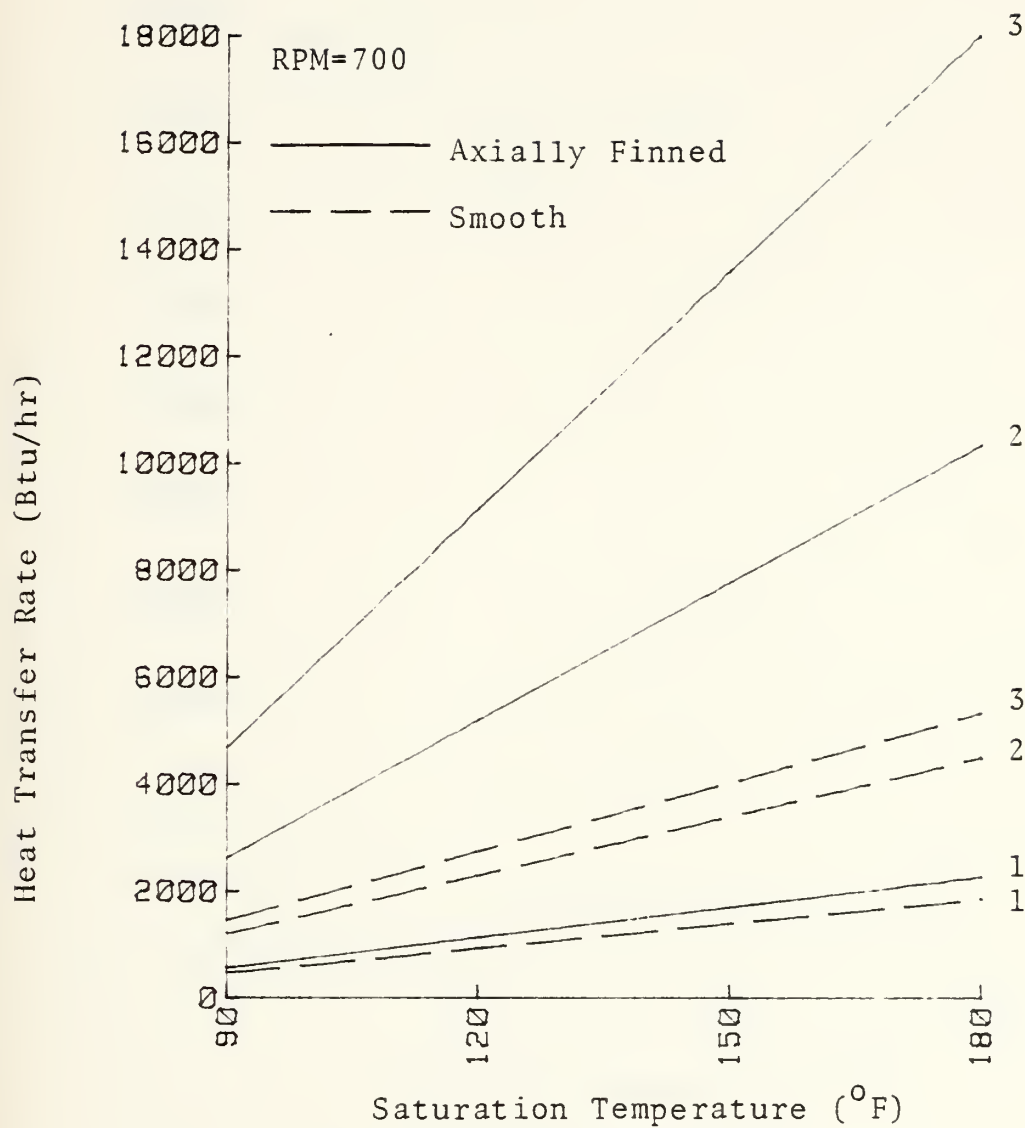
1. $h_{ext} = 100$ Btu/hr-ft²-°F
2. $h_{ext} = 500$ Btu/hr-ft²-°F
3. $h_{ext} = 1000$ Btu/hr-ft²-°F

Figure 18. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Stainless Steel Condensers at 1400 RPM



1. $h_{ext} = 100 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
2. $h_{ext} = 500 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
3. $h_{ext} = 1000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$

Figure 19. Heat Transfer Rate versus Saturation Temperature for Axially Finned Cylindrical and Tapered Stainless Steel Condensers at 2800 RPM



1. $h_{ext} = 100 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
2. $h_{ext} = 500 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
3. $h_{ext} = 1000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$

Figure 20. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Copper Cylindrical Condensers at 700 RPM

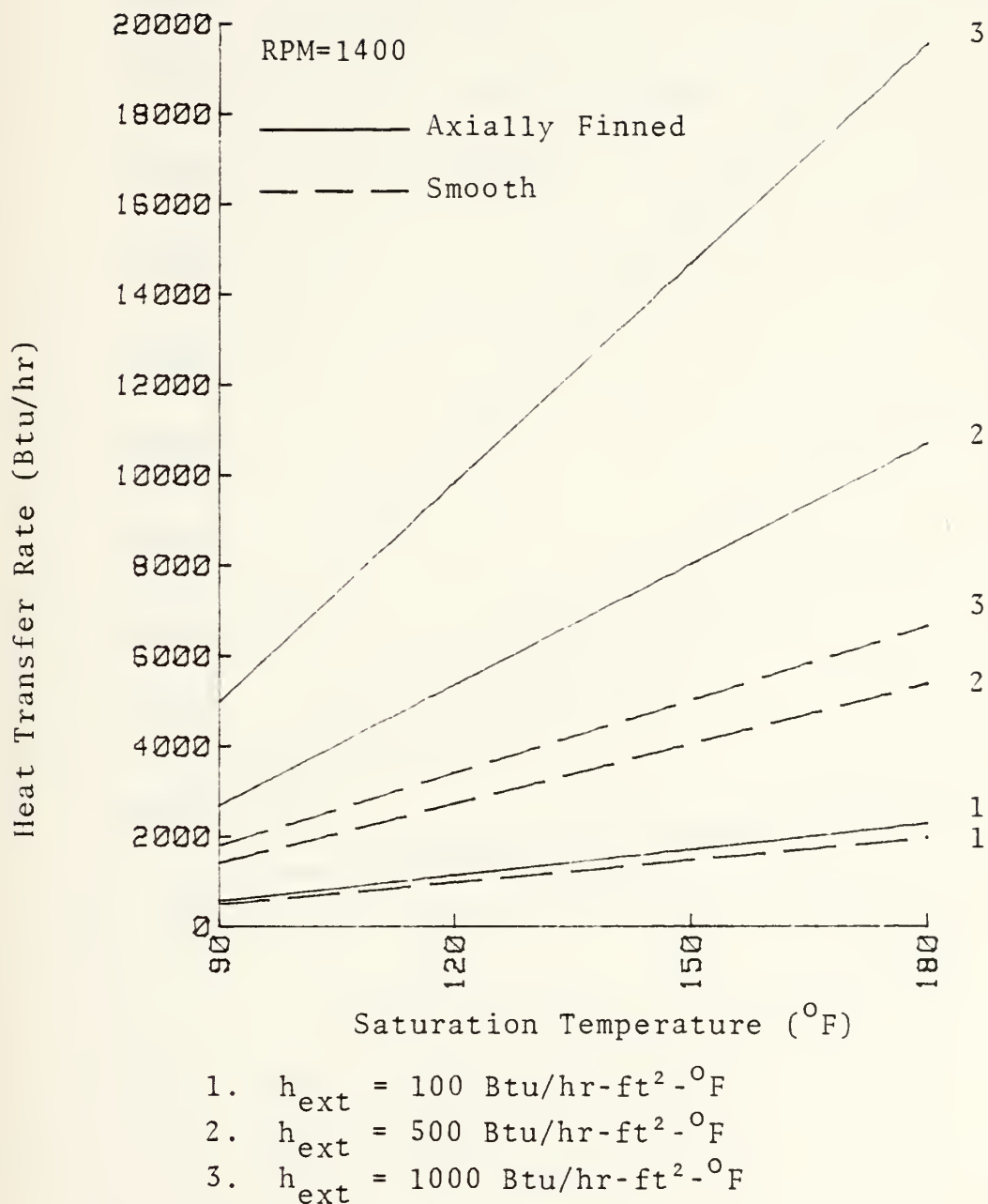
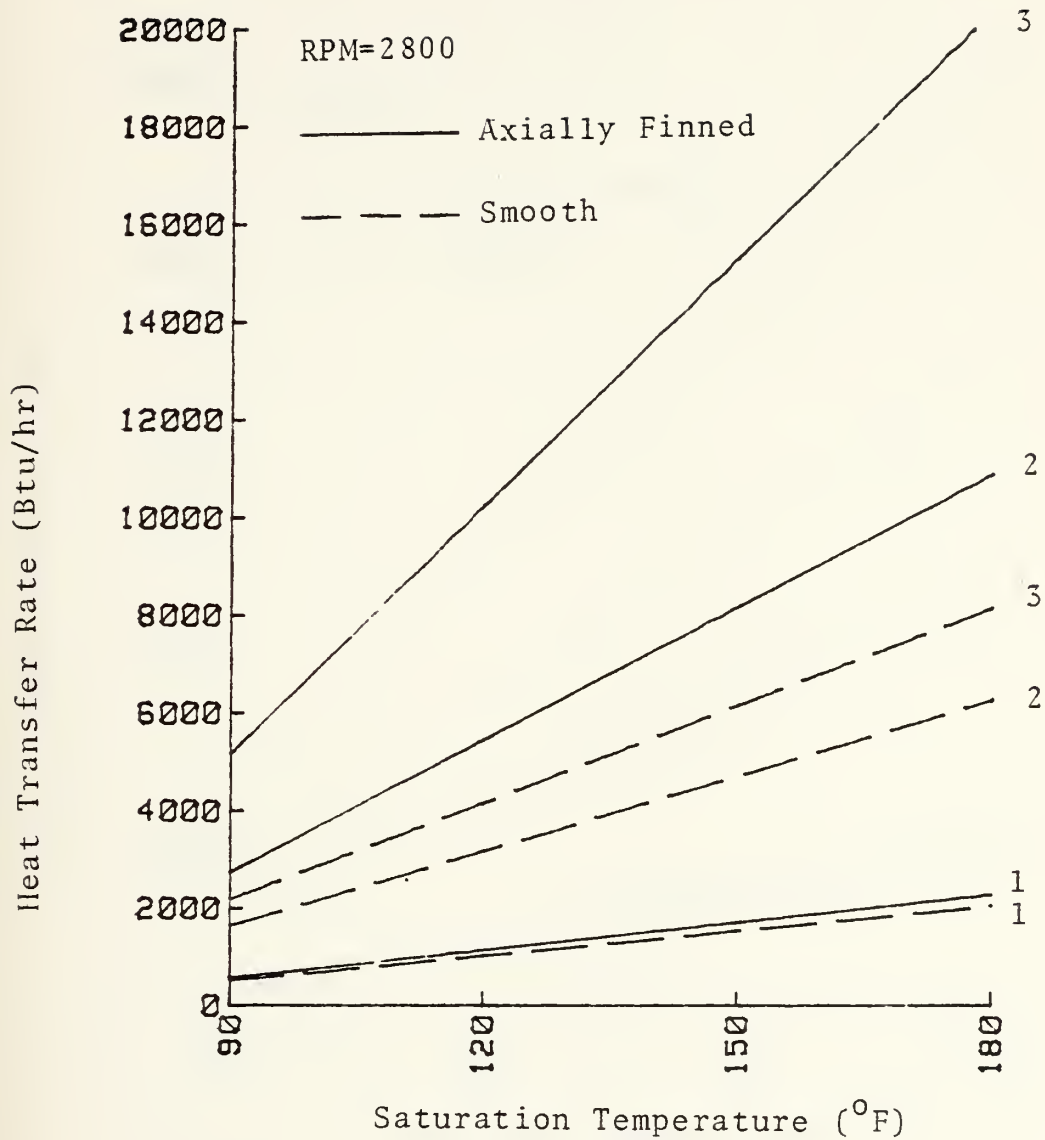
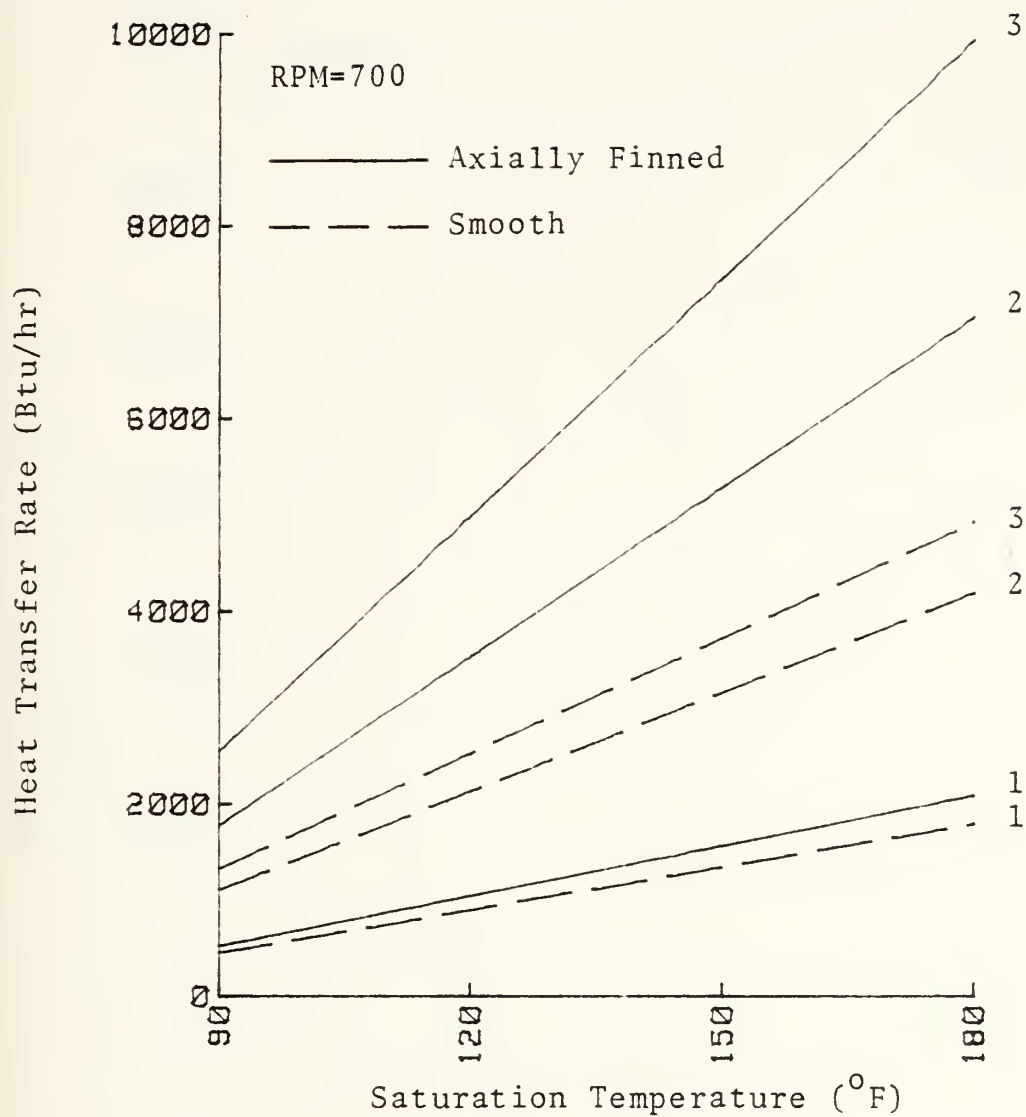


Figure 21. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Copper Cylindrical Condensers at 1400 RPM



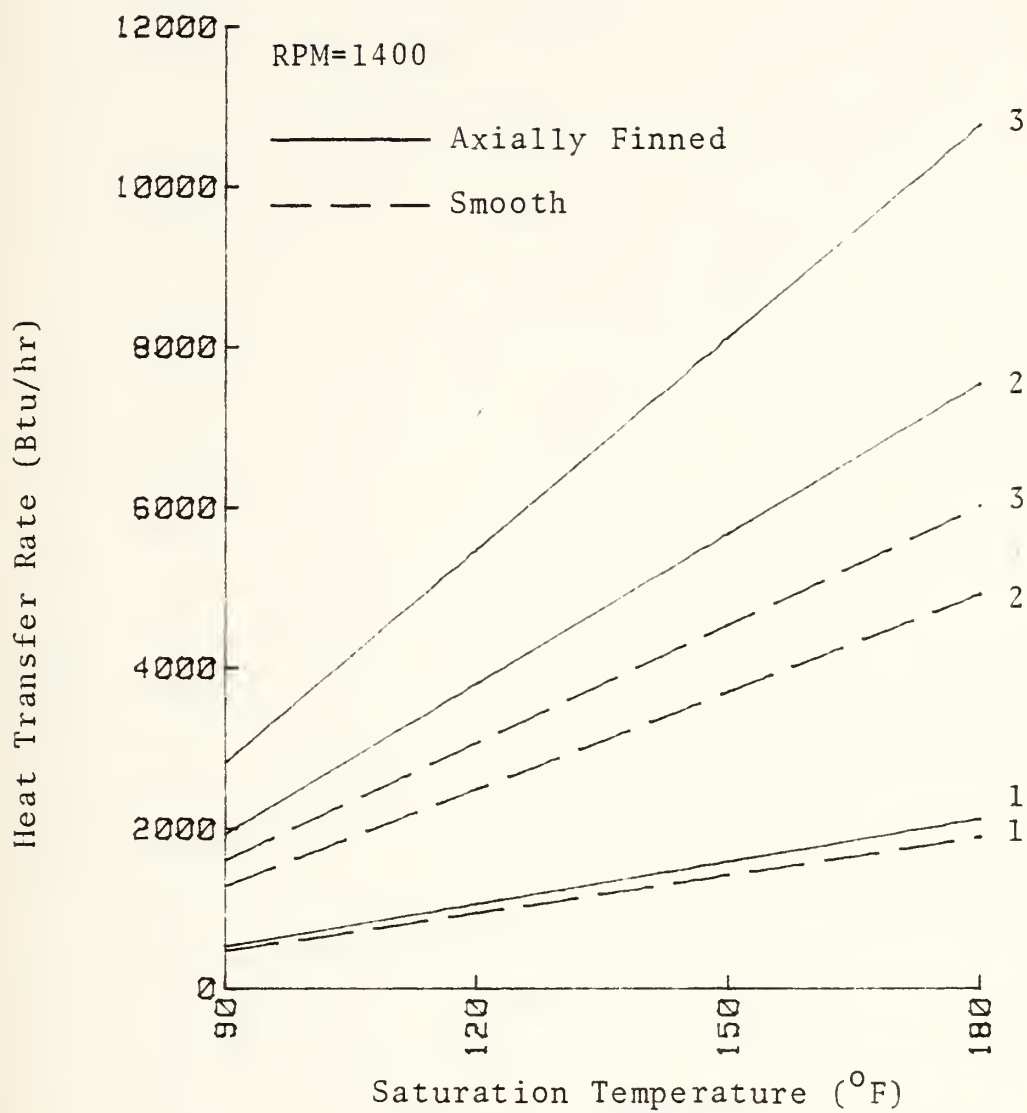
1. $h_{ext} = 100 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
2. $h_{ext} = 500 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
3. $h_{ext} = 1000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$

Figure 22. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Copper Cylindrical Condensers at 2800 RPM



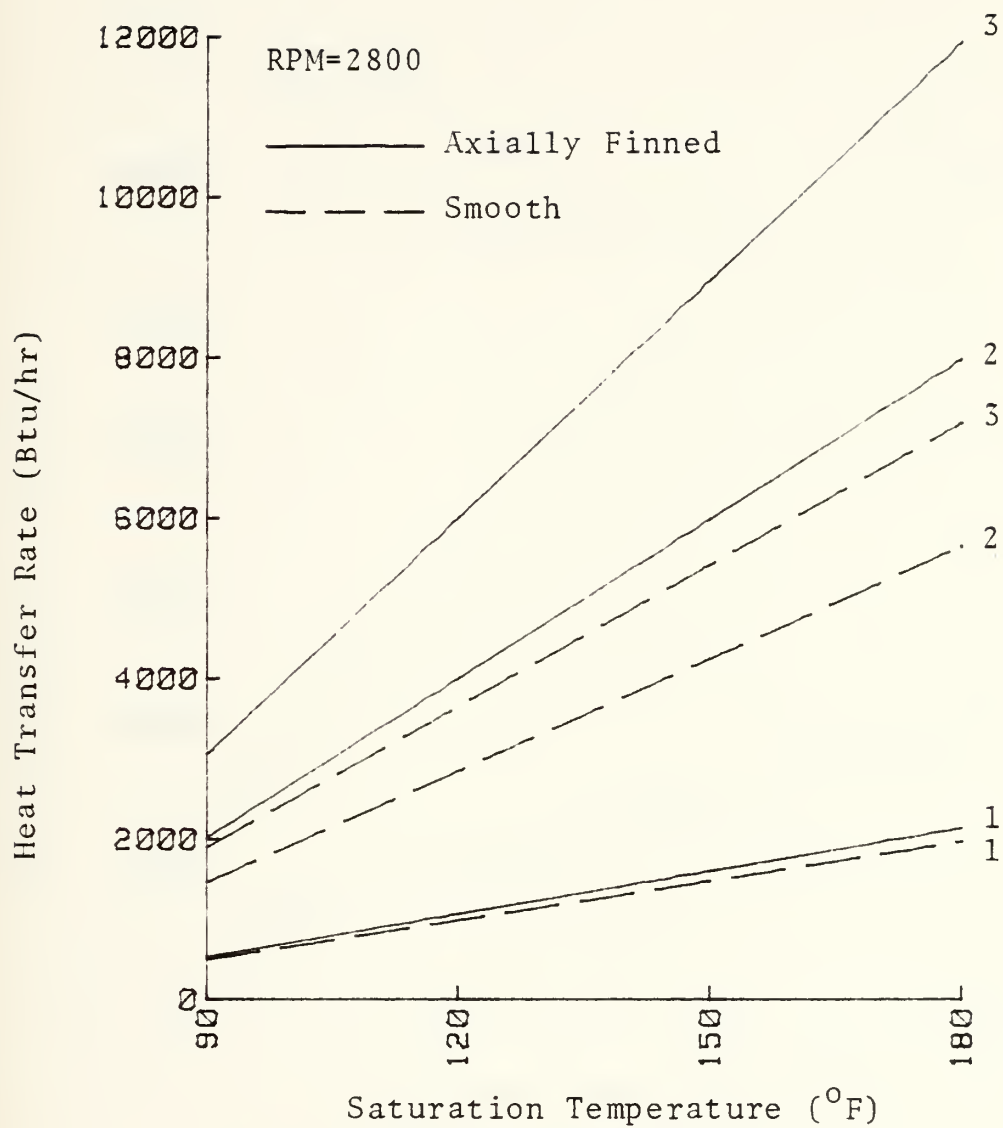
1. $h_{ext} = 100 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
2. $h_{ext} = 500 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
3. $h_{ext} = 1000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$

Figure 23. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Stainless Steel Cylindrical Condensers at 700 RPM



1. $h_{ext} = 100 \text{ Btu/hr-ft}^2\text{-}^{\circ}\text{F}$
2. $h_{ext} = 500 \text{ Btu/hr-ft}^2\text{-}^{\circ}\text{F}$
3. $h_{ext} = 1000 \text{ Btu/hr-ft}^2\text{-}^{\circ}\text{F}$

Figure 24. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Stainless Steel Cylindrical Condensers at 1400 RPM



1. $h_{ext} = 100 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
2. $h_{ext} = 500 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
3. $h_{ext} = 1000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$

Figure 25. Heat Transfer Rate versus Saturation Temperature for Smooth and Axially Finned Stainless Steel Cylindrical Condensers at 2800 RPM

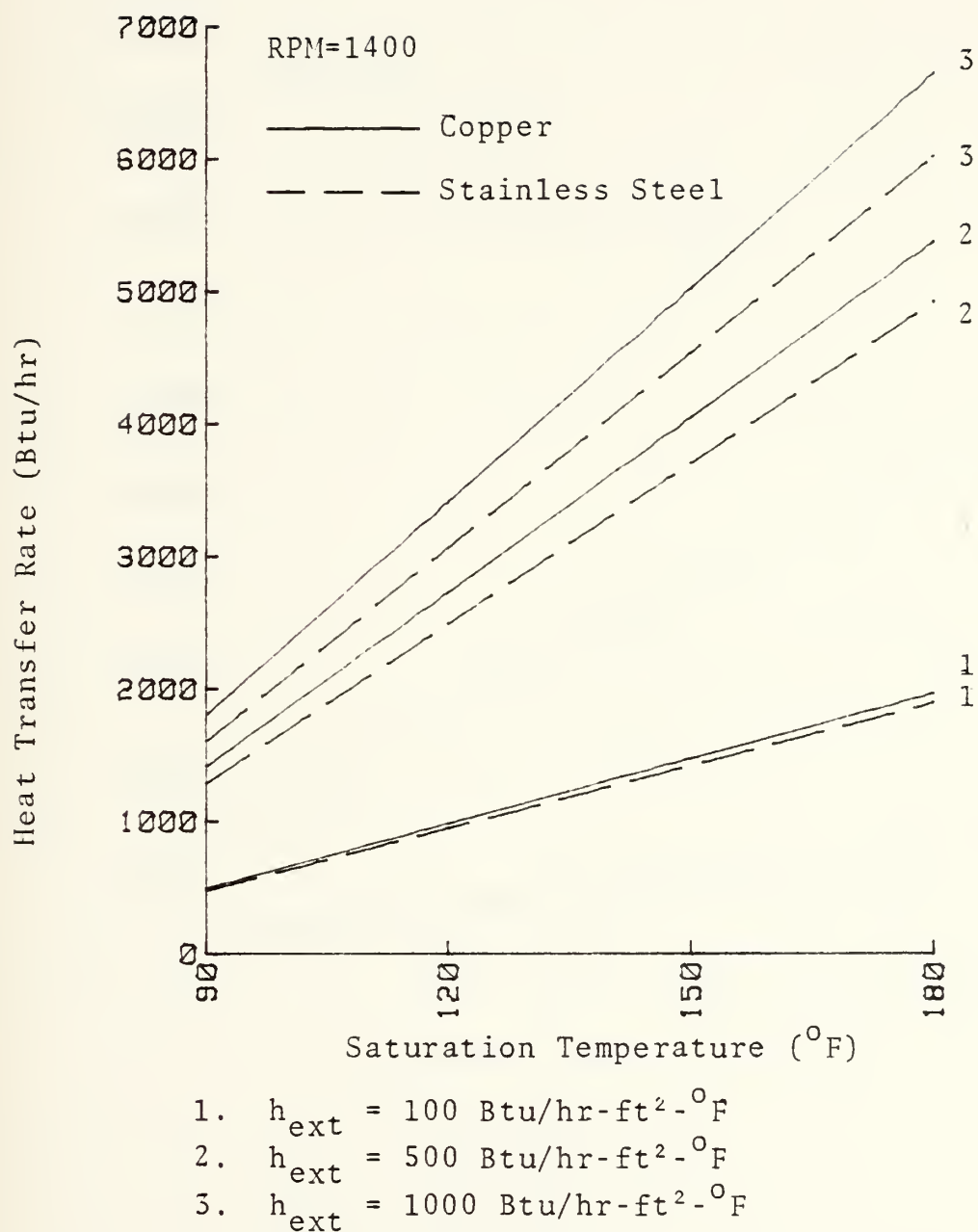
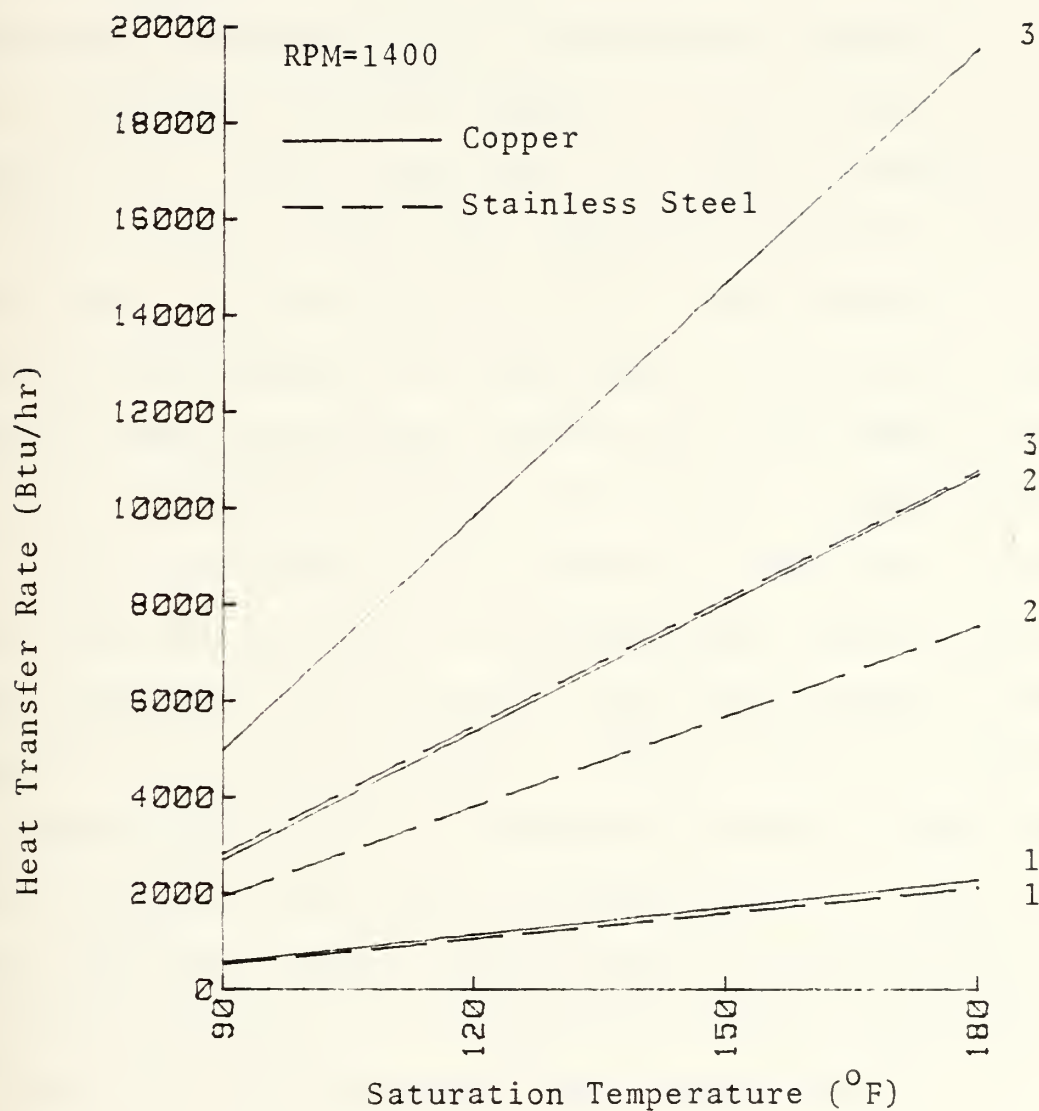


Figure 26. Heat Transfer Rate versus Saturation Temperature for Copper and Stainless Steel Smooth Cylindrical Condensers at 1400 RPM



1. $h_{ext} = 100 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
2. $h_{ext} = 500 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$
3. $h_{ext} = 1000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$

Figure 27. Heat Transfer Rate versus Saturation Temperature for Copper and Stainless Steel Axially Finned Cylindrical Condensers at 1400 RPM

V. CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

The heat transfer rate of a cylindrical condenser is less than an equivalent tapered condenser. This decrease in heat transfer rate is most significant in a smooth condenser, where, depending on the external heat transfer coefficient and rotational speed, can be as great as 45%. This decrease in heat transfer rate becomes less significant for an axially finned condenser where the average decrease, for the range of heat transfer coefficients examined, was 13% for a copper condenser and 18% for a stainless steel condenser. When such factors as the cost and difficulty in manufacturing tapered axially finned condensers are considered, this 13% decrease in heat transfer rate becomes tolerable. From a practical standpoint, development and analysis of cylindrical axially finned condensers should be encouraged by these results. If environmental conditions permit, copper should be preferred over stainless steel due to its exceptionally high thermal conductivity and resulting higher heat transfer rate.

B. RECOMMENDATIONS

- 1) Build and experimentally test both smooth and axially finned cylindrical condensers to obtain experimental data for comparison with results of this analysis.

2) Develop models for rectangular and trapezoidal fin profiles with non-adiabatic tips and incorporate into code.

3) Experimentally test axially finned cylindrical condensers with rectangular and trapezoidal fin profiles to obtain data for comparison with theoretical results.

APPENDIX A
FILM PROFILE FINITE ELEMENT SOLUTION

A. SMOOTH CONDENSER

The analysis of the film profile in a smooth cylindrical condenser developed in Chapter II resulted in the following ordinary, nonlinear, second order, differential equation:

$$\delta^* \frac{d}{dx} \left[\frac{d\delta^*}{dx} \delta^{*3} \right] = - \frac{3k_f(T_{sat} - T_w)\mu}{\rho_f^2 \omega^2 r \bar{h}_{fg}} \quad (\text{eqn A.1})$$

This equation can be rearranged and expanded to yield:

$$\delta^{*4} \frac{d^2 \delta^*}{dx^2} + 3\delta^{*3} \left(\frac{d\delta^*}{dx} \right)^2 = -K \quad (\text{eqn A.2})$$

$$\text{where } K = \frac{3k_f(T_{sat} - T_u)\mu}{\rho_f^2 \omega^2 r \bar{h}_{fg}}$$

The statement of the problem for the formulation of the Finite Element Method is:

$$\delta^{*4} \frac{d^2 \delta^*}{dx^2} + 3\delta^{*3} \left(\frac{d\delta^*}{dx} \right)^2 = -K \quad (\text{eqn A.3})$$

with the following boundary conditions:

a) at $x = 0$, $\delta^* = \delta_{\max}^*$

b) at $x = 0$, $d\delta^*/dx = 0$

The domain (length of the condenser) is divided into elements of length Δx with the exception of the first and final elements which have length $\Delta x/2$, where Δx is the length of the condenser divided by the number of axial increments (NDIV). A system nodal point is located at each end of an element. Thus, system nodal point 1 is located at $x=0$ and the last system nodal point, which is equal to the number of elements plus 1 is located at the overfall into the evaporator, $x=L$. All internal nodal points are located at a position corresponding to the midpoint of the axial increment.

Define the approximate value of the film thickness in the following manner:

$$\bar{\delta} \approx \delta_n^* (x) = \sum_1^h G_i d_i = G^T d \quad (\text{eqn A.4})$$

where $\bar{\delta}$ = the approximate value of the film thickness (δ^*).

G_i = the global basis functions.

n = the number of system nodal points.

d = the solution vector.

On an element level, equation (A.4) becomes:

$$\bar{\delta}_e \approx \bar{\delta}_e(x) = \sum_{i=1}^4 g_i d_{i_e} \quad (\text{eqn A.5})$$

where g_i = the local basis functions.

Define two degrees of freedom at each nodal point, i.e., both $\bar{\delta}$ and $d\bar{\delta}/dx$ are continuous. Thus, the local basis functions used in the finite element solution are:

$$g_1 = 1 - \frac{3\zeta^2}{\ell^2} + \frac{2\zeta^3}{\ell^3} \quad (\text{eqn A.6})$$

$$g_2 = \zeta - \frac{2\zeta^2}{\ell} + \frac{\zeta^3}{\ell^2} \quad (\text{eqn A.7})$$

$$g_3 = \frac{3\zeta^2}{\ell^2} - \frac{2\zeta^3}{\ell^3} \quad (\text{eqn A.8})$$

$$g_4 = -\frac{\zeta^2}{\ell} + \frac{\zeta^3}{\ell^2} \quad (\text{eqn A.9})$$

where ℓ = the length of an element.

g_1 and g_3 = magnitude basis functions.

g_2 and g_4 = slope basis functions.

Substituting the approximate film thickness $\bar{\delta}_e$ into the differential equation (A.3) results in:

$$\bar{\delta}_e^4 \frac{d^2 \bar{\delta}_e}{dx^2} + 3\bar{\delta}_e^3 \left(\frac{d\bar{\delta}_e}{dx} \right)^2 = -K_e \quad (\text{eqn A.10})$$

To remove the "nonlinearity" from the problem, equation (A.10) is modified in the following manner:

$$\eta^4 \frac{d^2 \bar{\delta}_e}{dx^2} + 3\eta^3 \eta' \frac{d\bar{\delta}_e}{dx} = -K_e \quad (\text{eqn A.11})$$

η is defined as the approximate value of the film thickness from the previous iteration. In like manner, η' is defined as the approximate value of the rate of change of film thickness with respect to x from the previous iteration.

Forming the residual of equation (A.11) yields:

$$R_n = \eta^4 (G^{T''} \underline{d}) + 3\eta^3 \eta' (G^{T'} \underline{d}) + K \quad (\text{eqn A.12})$$

Invoking the Galerkin criterion for the determination of the solution vector d , i.e.

$$\int_{\Omega} G_i R_n dx = 0 \quad i = 1, 2, 3, \dots, n \quad (\text{eqn A.13})$$

yields:

$$\eta^4 \int_{\Omega} G G^{T''} d dx + 3\eta^3 \eta' \int_{\Omega} G G^{T'} d dx + K \int_{\Omega} G d dx = 0 \quad (\text{eqn A.14})$$

Each of the integrals in equation (A.14) are defined in the following manner:

$$\int_{\Omega} G G^{T''} d dx = \sum_1^n g g^{T''} d_e dx \quad (\text{eqn A.15})$$

Or on an elemental level:

$$\int g g^{T''} d_e dx = \int_{\ell} \begin{Bmatrix} g_1 \\ g_2 \\ g_3 \\ g_4 \end{Bmatrix} \begin{matrix} < g_1'' & g_2'' & g_3'' & g_4'' > \\ & & & & \end{matrix} d_e dx \quad (\text{eqn A.16})$$

This integration results in the following 4x4 local elemental A matrix for any element:

$$[A]_e = \begin{bmatrix} \frac{-6}{\ell} & \frac{-11}{10} & \frac{6}{5\ell} & \frac{-1}{10} \\ \frac{-1}{10} & \frac{-2\ell}{15} & \frac{1}{10} & \frac{\ell}{30} \\ \frac{6}{5\ell} & \frac{1}{10} & \frac{-6}{5\ell} & \frac{11}{10} \\ \frac{-1}{10} & \frac{\ell}{10} & \frac{1}{10} & \frac{-2\ell}{15} \end{bmatrix}$$

In a similar manner, let

$$\int \tilde{G} G^{T'} d\tilde{x} = \sum_1^n \int_{\ell} \tilde{g} g^{T'} d_e dx \quad (\text{eqn A.17})$$

and on an elemental level:

$$\int_{\ell} \tilde{g} g^T d_e dx = \int_{\ell} \begin{Bmatrix} g_1 \\ g_2 \\ g_3 \\ g_4 \end{Bmatrix} \langle g_1' \ g_2' \ g_3' \ g_4' \rangle d_e dx \quad (\text{eqn A.18})$$

This integration results in the following 4x4 local elemental B matrix for any element:

$$[B]_e = \begin{bmatrix} \frac{-1}{2} & \frac{\ell}{10} & \frac{1}{2} & \frac{-\ell}{10} \\ \frac{-\ell}{10} & 0 & \frac{\ell}{10} & \frac{-\ell^2}{60} \\ \frac{-1}{2} & \frac{-\ell}{10} & \frac{1}{2} & \frac{\ell}{10} \\ \frac{\ell}{10} & \frac{\ell^2}{60} & \frac{-\ell}{10} & 0 \end{bmatrix}$$

Lastly, let

$$\int \underline{G} dx = \sum_{1}^n \int_{\ell} \underline{g} dx \quad (\text{eqn A.19})$$

or, on an elemental level this becomes:

$$\int_{\ell} \underline{g} dx = \int_{\ell} \begin{Bmatrix} g_1 \\ g_2 \\ g_3 \\ g_4 \end{Bmatrix} dx \quad (\text{eqn A.20})$$

This integration results in the following local elemental column F vector:

$$[F]_e = \begin{Bmatrix} \frac{\ell}{2} \\ \frac{\ell^2}{12} \\ \frac{\ell}{2} \\ \frac{\ell^2}{12} \end{Bmatrix}$$

Thus, for a given element, equation (A.12) becomes:

$$[\eta_e]^4 [A]_e + 3 \eta_e^3 \eta_e' [B]_e] \underline{d}_e = -K_e \underline{F}_e \quad (\text{eqn A.21})$$

As mentioned above, η and η' are the approximate values of δ^* and $d\delta^*/dx$ respectively from the previous iteration. For the initial iteration, η is set equal to δ_{\max}^* and η' is set equal to 0.

Each elemental matrix is placed in a global system [A] matrix with the location in the matrix based upon the local/global

nodal point correspondence. For example, for element Nr. 2 the nodal points corresponding to local nodal points 1,2,3, and 4 are global nodal points 3,4,5, and 6 respectively. Therefore, the sum of the two 4x4 elemental matrices ([A] and [B]) and multiplying constants will form a single 4x4 local matrix. Element (1,1) of this local matrix will be placed in the global A matrix location (3,3) and element (4,4) of the local elemental matrix will be placed in the global A matrix location (6,6). Ultimately, the following global system would be assembled:

$$[A]_{mxm} \cdot \underline{d} = \underline{F} \quad (\text{eqn A.22})$$

Note that the global A matrix would be an mxm size matrix where m is equal to twice the number of system nodal points to account for the two degrees of freedom at each nodal point. In a similar fashion, \underline{d} and \underline{F} would be column vectors of size m.

Once the system is assembled the boundary conditions are applied. This is done in the following manner: For boundary condition (a), A(1,1) is set equal to 1.0 and the remaining elements in the first row, i.e. A (1,i) i=2,3,4,...m are set equal to 0.0. Then F(1) is set equal to δ_{\max}^* . δ_{\max}^* is initially determined by a relationship developed by Leppert and Nimmo [Refs. 8 and 9]. This establishes the value of d(1) as δ_{\max}^* . In a similar manner, boundary condition (b) is applied by setting A(2,2) equal to 1.0 and all other second row elements of the global A matrix are set equal to 0.0. Then, F(2) is set

equal to 0.0. This establishes the solution vector $d(2)$ as 0.0. Note that $d(2)$ corresponds to the slope at the first nodal point.

One additional boundary condition is required; this is the value of the film thickness at $x=L$. This boundary condition is necessary to completely specify the problem. The value of the film thickness at the overfall may take on any value depending on the geometry at the overfall. In the case of this analysis, this value was taken as $0.25 \cdot \delta_{\max}^*$. This value was chosen for the following reason: Leppert and Nimmo [Refs. 8 and 9], in a similar analysis for laminar film condensation on a horizontal surface derived an analytical solution to equation (A.1), assuming a constant surface temperature. They found the film profiles for δ^* with the overfall value less than $0.40 \cdot \delta_{\max}^*$ were essentially constant and thus any value of δ^* at the overfall less than $0.40 \cdot \delta_{\max}^*$ would result in the same profile. In the verification of the finite element solution, not only was this found to be the case, but it was also found that the heat transfer rate was relatively insensitive to the shape of the film profile. In fact, it was found that the film thickness at the overfall could be increased to a value as great as $0.90 \cdot \delta_{\max}^*$ and the resulting variation in heat transfer rate was only 10%. This being the case, the value of $0.25 \cdot \delta_{\max}^*$ was arbitrarily chosen for δ_{\min} .

The third boundary condition was applied by setting $A(m-1,i)=0.0$ where $i=1,2,3\dots m$. Then, the global matrix element $A(m-1,m-1)$ was set equal to 1.0. Finally, $F(m-1)$ was set equal to $0.25 \cdot \delta_{\max}^*$.

Once all three boundary conditions were applied, the system given by equation (A.22) was solved for \underline{d} . The values of the approximate film thickness, i.e., $d(i)$, $i=1,3,5,\dots,m-1$ are then compared to the values of film thickness from the previous iteration. If the relative difference is less than or equal to a specified convergence criterion (i.e., 0.0001) at all nodal points, convergence is met and the latest values of d are the solution values of δ^* .

If convergence is not met, the values of d are saved for the next iteration where they are used to determine η and η' as discussed above. This iterative process is continued until convergence is met or until a maximum number of iterations have occurred.

B. AXIALLY FINNED CONDENSER

The finite element solution for the film profile in an axially finned cylindrical condenser is very similar to that of a smooth cylindrical condenser. For this reason, only the variations in the development will be addressed. From Chapter II, the differential equation for mass flow rate in an axially finned cylindrical condenser is given by:

$$\delta^* \frac{d}{dx} \left[\frac{d\delta^*}{dx} (\epsilon \delta^{*3} + \delta^{*4} \tan \alpha) \right] = - \frac{3k_f(T_{sat} - T_w)\mu\epsilon}{\rho_f^2 \omega^2 r \bar{h}_{fg}}$$

$$- 2\delta^* \cos \alpha \left[\frac{4k_f(T_{sat} - T_{avg})\mu z^*}{\rho_f^2 \omega^2 r \bar{h}_{fg} \cos \alpha} \right]^{3/4} \quad (\text{eqn A.23})$$

This equation can be rearranged and expanded to yield:

$$\begin{aligned} \frac{d^2 \delta^*}{dx} (\epsilon \delta^{*4} + \delta^{*5} \tan \alpha) + \frac{d \delta^*}{dx} (3 \epsilon \delta^{*3} \frac{d \delta^*}{dx} + 4 \delta^{*4} \tan \alpha \frac{d \delta^*}{dx}) \\ = -K_1 - K_2 \delta^* \end{aligned} \quad (\text{eqn A.24})$$

$$\text{where } K_1 = \frac{3k_f(T_{\text{sat}} - T_w)\mu\epsilon}{\rho_f^2 \omega^2 r h_{fg}}$$

$$K_2 = 2 \cos \alpha \left[\frac{4k_f(T_{\text{sat}} - T_{\text{avg}})\mu z^*}{\rho_f^2 \omega^2 r h_{fg} \cos \alpha} \right]^{3/4}$$

Substituting equation (A.5) into equation (A.24) results in:

$$\begin{aligned} \frac{d^2 \bar{\delta}_e}{dx^2} (\epsilon \bar{\delta}_e^4 + \bar{\delta}_e^5 \tan \alpha) + \frac{d \bar{\delta}_e}{dx} (3 \epsilon \bar{\delta}_e^3 \frac{d \bar{\delta}_e}{dx} + 4 \bar{\delta}_e^4 \tan \alpha \frac{d \bar{\delta}_e}{dx}) \\ = -K_1 - K_2 \bar{\delta}_e \end{aligned} \quad (\text{eqn A.25})$$

To remove the "nonlinearity" from the problem, equation (A.25) is modified in the following manner"

$$\begin{aligned} \frac{d^2 \bar{\delta}_e}{dx^2} (\epsilon \gamma^4 + \gamma^5 \tan \alpha) + \frac{d \bar{\delta}_e}{dx} (3 \epsilon \gamma^3 \gamma' + 4 \gamma^4 \gamma' \tan \alpha) \\ = -K_1 - K_2 \gamma \end{aligned} \quad (\text{eqn A.26})$$

Here, γ and γ' are defined by the following relationships:

$$\gamma = \bar{\delta}_i + R^*(\bar{\delta}_i - \bar{\delta}_{i-1}) \quad (\text{eqn A.27})$$

$$\gamma' = \bar{\delta}'_i + R^*(\bar{\delta}'_i - \bar{\delta}'_{i-1}) \quad (\text{eqn A.28})$$

where $\bar{\delta}_i$ = approximate value of the film thickness for the present iteration.

$\bar{\delta}_{i-1}$ = approximate value of film thickness from the previous iteration.

$\bar{\delta}'_i$ = approximate value of the derivative of the film thickness for the present iteration

$\bar{\delta}'_{i-1}$ = approximate value of the derivative of the film thickness from the previous iteration.

R = relaxation factor.

These two variables, γ and γ' , are in actuality, adjusted approximation of film thickness and derivative of film thickness respectively. This adjustment is required in order to converge to a solution.

The finite element solution is now identical to that of the smooth condenser, that is, the residual is formed, the Galerkin criterion is invoked, and identical 4x4 local matrices are derived. Finally, the equivalent of equation (A.21) is formed.

$$\begin{aligned} & [(\epsilon\gamma_e^4 + \gamma_e^5 \tan\alpha) [A]_e + (3\epsilon\gamma_e^3 \gamma'_e + \gamma_e^4 \gamma'_e \tan\alpha) [B]_e] \cdot \underline{d}_e \\ & = (- K_{1e} - K_{2e} \gamma_e) \underline{F}_e \end{aligned} \quad (\text{eqn A.29})$$

Notice that this equation has two forcing terms. The additional term $(K_2\gamma)$ resulted from the nonlinearization of the problem.

Just as in the smooth condenser film profile solution, the global system given by equation (A.22) is formed, the boundary conditions applied and the system solved for a solution vector \underline{d} . The iterative process is continued until convergence is met. With each iteration, γ and γ' are updated and used for the next iteration. When convergence is met, the latest values of $d(i)$, $i=1,3,5,\dots,m-1$ are the solution values of δ^* .

APPENDIX B
USER'S MANUAL

This appendix describes the data cards required to use the computer code.

The data is divided into "blocks" for convenience. Each page of this user's guide is a separate block. For each block, a general description, the required format, and appropriate comments are provided.

It is imperative that input data be consistent or errors will result. For example, if a smooth geometry is being analyzed, the finite element parameters must also result in a smooth model. In addition, all data fields must be filled with an input value, even if that value is not needed for the analysis. For example, in a smooth analysis, no fin half angle is required for the calculations; however, a value of the correct format must be provided in the fin half angle field or an INPUT/OUTPUT error will result.

DATA BLOCK A

DESCRIPTION: FINITE ELEMENT PARAMETERS

FORMAT: 8I5

1	2	3	4	5	6	7	8
-----	-----	-----	-----	-----	-----	-----	-----
NDIV	NCMREC	NCMTRI	NRWFIN	NRWTRF	NCMTRF	NCOL	NPRNT
-----	-----	-----	-----	-----	-----	-----	-----

FIELD CONTENTS

- 1 NDIV---Number of axial increments. Must be less than or equal to 50.
- 2 NCMREC-Number of columns of finite elements in the rectangular section of fin. May be equal to 0 if fin is triangular only.
- 3 NCMTRI-Number of columns of finite elements in triangular section of fin. May be equal to 0 if only rectangular fin.
- 4 NRWFIN-Number of rows of finite elements in the fin section of model. May be equal to 0 if a smooth condenser.
Note: If triangular or trapezoidal fin, NRWFIN must equal NCMTRI.
- 5 NRWTRF-Number of rows of finite elements in wall section of the model.
- 6 NCMTRF-Number of columns of finite elements in the trough section of finned model. Set equal to 0 if a smooth condenser.
- 7 NCOL---Total number of columns of finite elements. Must be equal to NCMREC+NCMTRI+NCMTRF for a finned model.
- 8 NPRNT--Print control number.
If equal to 1 correspondence table and major elements of finite element model will be printed out.
If equal to 0, output will be suppressed.

DATA BLOCK B

DESCRIPTION: FLUID, FIN MATERIAL SELECTION PARAMETERS

FORMAT: 2I5

1	2	3	4	5	6	7	8
---	---	---	---	---	---	---	---

IFLUID	IFIN
--------	------

<u>FIELD</u>	<u>CONTENTS</u>
--------------	-----------------

- | | |
|---|--|
| 1 | IFLUID-If equal to 0, working fluid is water.
If equal to 1, working fluid is freon. |
| 2 | IFIN---If equal to 0, condenser wall material is
copper.
If equal to 1, condenser wall material is
stainless steel. |

DATA BLOCK C

DESCRIPTION: CONDENSER GEOMETRY

FORMAT: 6G10.5

1 2 3 4 5 6 7 8

CLI REASEI THICKI BFINI CANGL FNWTHI

FIELD CONTENTS

1 CLI----Condenser length (inches).
2 RBASEI-Inside radius to wall of condenser at condenser
end (inches).
3 THICKI-Wall thickness (inches).
4 BFINI--Fin height (inches). Must be set equal to
0.0 if smooth condenser.
5 CANGL--Condenser half angle for tapered condenser
(degrees). Must be set equal to 0.0 for
cylindrical condenser.
6 FNWTHI-Width of rectangular portion of trapezoidal
or rectangular fin (inches). If triangular
fin, set equal to 0.0.

DATA BLOCK D

DESCRIPTION: INTERNAL FIN GEOMETRY

FORMAT: 2G10.5

1	2	3	4	5	6	7	8
---	---	---	---	---	---	---	---

FANGL ETOEO

<u>FIELD</u>	<u>CONTENTS</u>
--------------	-----------------

1	FANGL--Fin half angle (degrees). Set equal to 0.0 for smooth condenser or rectangular fin.
2	ETEO--Ratio of trough width to fin base width. Determines spacing between fins.

DATA BLOCK E

DESCRIPTION: CONVERGENCE CRITERIA

FORMAT: 2G10.5

1 2 3 4 5 6 7 8

CRIT CRITDL

FIELD CONTENTS

- 1 CRIT---Temperature convergence criterion. Used to
determine solution of two dimensional steady
state heat conduction problem.
- 2 CRITDL-Mass flow convergence criterion. Used only
in cylindrical condenser analysis for mass
flow convergence test.

DATA BLOCK F

DESCRIPTION: OPERATING PARAMETERS

FORMAT: 5G10.5

1	2	3	4	5	6	7	8

RPM	HINF	TINTL	TSAT	TINF			

<u>FIELD</u>	<u>CONTENTS</u>
1	RPM---Rotational speed (revolutions per minute).
2	HINF---External heat transfer coefficient (Btu/hr-ft -deg F).
3	TINTL--Initial temperature estimate (degrees F).
4	TSAT---Saturation temperature (degrees F).
5	TINF---Ambient temperature (degrees F).

DATA BLOCK G

DESCRIPTION: OUTPUT PRINT CONTROL

FORMAT: 4I5

1	2	3	4	5	6	7	8
---	---	---	---	---	---	---	---

IUNITS NFLAG1 NFLAG2 NFLAG3

FIELD	CONTENTS
-------	----------

- | | |
|---|--|
| 1 | IUNITS-Output units control number.
If IUNITS = 0, calculated results will be provided in English units.
If IUNITS = 1, input parameters will be repeated in SI units and calculated results will be provided in SI units.
If IUNITS = 2, input parameters will be repeated in SI units and output results will be provided in both English and SI units. |
| 2 | NFLAG1-The first axial increment at which the parameters listed under remarks will be provided as output. |
| 3 | NFLAG2-The final increment at which the parameters listed under remarks will be provided. |
| 4 | NFLAG3-The step change in increments between NFLAG1. |

REMARKS

- 1) No matter what value of IUNITS is used, input parameters will always first appear in English units.
- 2) For increments indicated by NFLAG values, the following parameters will appear: a) x-coordinate, y-coordinate and temperature at each nodal point, b) length of element, heat transfer coefficient and heat rate per unit length for each convective boundary element.
- 3) As a minimum, the values of 1, 1, 1 must be provided as input for NFLAG1, NFLAG2, and NFLAG3 respectively.

DATA BLOCK H

DESCRIPTION: TAPERED CONDENSER SOLUTION METHOD

FORMAT: 115

1	2	3	4	5	6	7	8
---	---	---	---	---	---	---	---

NSOLVE

FIELD CONTENTS

- 1 NSOLVE-Tapered solution control number.
 For tapered, axially finned condenser, set
 NSOLVE=1.
 For tapered smooth condenser, NSOLVE must be
 set to one of the following three values:
 Set NSOLVE = 2 if solution of film thickness is
 to be based on Ballback's [Ref. 1]
 equation.
 Set NSOLVE = 3 if solution of film thickness is
 to be based on Daniels and Al-Jumaily [Ref. 13]
 equation, neglecting drag terms.
 Set NSOLVE = 4 if solution of film thickness is
 to be based on Daniel's and Al-Jumaily [Ref.
 13] equation, with drag effects included.

REMARKS

- 1) NSOLVE is only used in tapered condenser analysis.

DATA BLOCK I

DESCRIPTION: CYLINDRICAL CONDENSER ANALYSIS PARAMETERS

FORMAT: 3I5,2GI5.10

1	2	3	4	5	6	7	8
---	---	---	---	---	---	---	---

NONCE	ITERMX	ITRPRT	RELAX	DELMAX			
-------	--------	--------	-------	--------	--	--	--

<u>FIELD</u>	<u>CONTENTS</u>
--------------	-----------------

- | | |
|---|--|
| 1 | NONCE--Single iteration parameter.
If NONCE = 1, only one iteration will be permitted.
If NONCE = 0, iterations will continue until convergence or maximum number of iterations is reached. |
| 2 | ITERMX-Maximum number of iterations permitted in analysis. |
| 3 | ITRPRT-Iteration print control parameter.
If ITRPRT = 1, mass flow convergence test results will be provided for each iteration.
If ITRPRT = 0, mass flow convergence test results will only be provided on final iteration. |
| 4 | RELAX--Relaxation variable used in finite element solution of cylindrical finned film profile. |
| 5 | DELMAX-Initial estimate of maximum film thickness used in solution of cylindrical finned film profile. |

REMARKS

- 1) Above parameters are only used in cylindrical condenser analysis.
- 2) Recommended value of RELAX is 0.80. It is sometimes necessary to adjust this value plus or minus 0.05 to reach film profile convergence at small film thickness values.
- 3) Input value of DELMAX is only used in cylindrical finned analysis. For the cylindrical smooth condenser, DELMAX is internally generated.

APPENDIX C

SOURCE CODE LISTING

```

IMPLICIT REAL*8(A-H,O-Z)
DIMENSION DIFF(200),TI(200),TAVG(100),A(200,50),F(
1200,1),TPLT(025,100),XPLT(025,100),YPLT(025,100),DLNGTH(50)
COMMON /GLOBEL/ BOA,BFINI,CANGL,CLI,ETOEO,FANGL,HINF,QBTOT,RBASEI,R
1PM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
COMMON /GLOB2/AFGVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
1ITDL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
2LUMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
3(100,50),QFNTOI,QINC(100),QINCSM(100),QSMTOI,QTFID,QTINC(100),QTO
4TAL(100),QX(100),R(100),RHO(100),SALFA,SLNGTH(100),SPHI,SURFAR,T(
5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TIRDF(100),THICK
6UF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NEXTLT,NEXTRT,NPFSY
7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRT,NPDIFF,NPFCNV(10),NPFSY
8M(10),NPMSBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMKEC,NRWREC,NCUL,NSOLVE
COMMON/DELI/RELAX,DELMAX,ITRPR
COMMON/RECT/FNWDTH,NFNTIP

```

***** INPUT MODE *****

PRINT HEADER FOR FINITE ELEMENT PARAMETERS

WRITE (6,74C)

INPUT MAJOR FINITE ELEMENT PARAMETERS

```

NDIV-----NUMBER OF INCREMENTS ALONG CONDENSED LENGTH OF FIN
NCMREC-----NUMBER OF COLUMNS IN RECTANGULAR PORTION OF FIN
(MAY BE EQUAL TO ZERO IF ONLY TRIANGULAR FIN)
NCMTRI-----NUMBER OF COLUMNS IN THE TRIANGULAR PORTION OF FIN
(MAY BE EQUAL TO ZERO IF ONLY RECTANGULAR FIN)
THE GEOMETRY IS SUCH THAT THE NUMBER OF COLUMNS IN
THE TRIANGULAR PORTION OF THE FIN WILL BE EQUAL
TO THE NUMBER OF ROWS IN THE FIN SECTION
NRWFIN-----NUMBER OF ROWS IN THE FIN SECTION-GEOMETRY IS SUCH
THAT THE NUMBER OF ROWS IN THE RECTANGULAR SECTION
WILL BE THE SAME AS THE NUMBER OF ROWS IN THE TRI-
ANGULAR SECTION OF FIN
NRWTRF-----NUMBER OF ROWS IN THE TROUGH SECTION
NCMTRF-----NUMBER OF COLUMNS IN THE FINITE ELEMENT MODEL
NCOL-----TOTAL NUMBER OF COLUMNS IN THE FINITE ELEMENT MODEL
NPRNT-----FINITE ELEMENT PRINT CONTROL NUMBER
IF NPRNT=0,ONLY FINITE ELEMENTS LISTED A-
BOVE WILL BE PROVIDED IN OUTPUT
IF NPRNT=1,NOT ONLY ABOVE PARTS AS WELL AS MAJOR ELE-
MENTS WITH CORRESPONDING NODAL POINTS AS WELL AS LISTED-USEFUL IN
TROUGH SHOOTING

```


INPUT SOLUTION METHOD VARIABLE FOR TAPERED SMOOTH CASE

NSOLVE-----SOLUTION METHOD VARIABLE
NSOLVE=1 IF FINNED TAPERED HEAT PIPE
NSOLVE=2 BALLBACK'S EQUATION FOR FILM THICKNESS IS
USED
NSOLVE=3 DANIEL'S AND AL-JUMAILY EQUATION, WITHOUT
DRAG IS USED TO DETERMINE FILM THICKNESS
NSOLVE=4 DANIEL'S AND AL-JUMAILY EQUATION, WITH DRAG
IS USED TO DETERMINE FILM THICKNESS

READ (5,720)NSOLVE

INPUT CYLINDRICAL ANALYSIS PARAMETERS

NONCE-----IF EQUAL TO 1, ONLY ONE ITERATION WILL BE ACCOMPLISHED
IF EQUAL TO 0, ITERATIONS WILL CONTINUE UNTIL MASS
FLOW RATE CONVERGENCE IS REACHED
ITRMX-----MAXIMUM NUMBER OF MASS FLOW ITERATIONS PERMITTED
ITRPT-----IF EQUAL TO 1, RESULTS OF MASS FLOW TEST WILL BE
PRINTED FOR EACH ITERATION
IF EQUAL TO 0, RESULTS OF MASS FLOW TEST WILL BE PRINT-
ED ONLY ON FINAL ITERATION
RELAX-----RELAXATION VARIABLE USED TO ADJUST APPROXIMATE SOLU-
TION OF FILM THICKNESS IN FINITE ELEMENT SOLUTION OF
FILM THICKNESS PROFILE
DELMAX-----MAXIMUM FILM THICKNESS INITIAL ESTIMATE. WILL BE AD-
AUTOMATICALLY ADJUSTED AFTER FIRST ITERATION. INPUT
VALUE IS ONLY USED IN CYLINDRICAL FINNED CONDENSER
ANALYSIS. FOR SMOOTH CYLINDRICAL CONDENSER ANALYSIS,
DELMAX IS A CALCULATED VALUE.

READ(5,730)NONCE,ITRMX,ITRPT,RELAX,DELMAX

***** EXECUTION MODE *****

***** BEGIN EXECUTION MODE *****

ESTABLISH CORRESPONDENCE BETWEEN NODAL POINTS AND ELEMENTS
AND DEFINE OTHER FINITE ELEMENT PARAMETERS

CALL CFRES (NPRNT, NBAN, NEXIM, NFINM)

CONVERT UNITS OF ALL DIMENSIONAL PARAMETERS FROM INCHES TO FEET
 CONVERT UNITS OF ANGLES FROM DEGREES TO RADIAN. CALCULATE ADDI-
 TIONAL CONDENSER GEOMETRIC VARIABLES.

```

CL-----CONDENSER LENGTH(FEET)
RBASE-----INSIDE RADIUS OF CONDENSER AT BASE (FEET)
R2-----AVERAGE CONDENSER RADIUS(FEET)
BFIN-----FIN HEIGHT(FEET)
FNWDTH-----WIDTH OF RECTANGULAR PORTION OF FIN(FEET)
              NOTE: MAY BE EQUAL TO ZERO IF ONLY TRIANGULAR FIN
PHI-----CONE HALF ANGLE OF CONDENSER (RADIAN)
SPHI-----SINE OF PHI
CPHI-----COSINE OF PHI
TPHI-----TANGENT OF PHI
DELX-----INCREMENT WIDTH(FEET)
CBASE-----BASE CIRCUMFERENCE(FEET)
REXIT-----CONDENSER RADIUS AT EXIT(EVAPORATOR SECTION)(FEET)
CEXIT-----BASE CIRCUMFERENCE AT EXIT(FEET)
THICK-----CONDENSER WALL THICKNESS AT TROUGH(FEET)
ALFA-----FIN HALF ANGLE(RADIAN)
SALFA-----SINE OF ALFA
CALFA-----COSINE OF ALFA
TALFA-----TANGENT OF ALFA
EZERO-----WIDTH OF FIN BASE(FEET)
EPSO-----TROUGH WIDTH(FEET)
ZFIN-----NUMBER OF FINS
SURFAR-----SURFACE AREA OF FIN AND TROUGH PER UNIT LENGTH(FT2)
EPSEX-----TROUGH WIDTH AT EXIT(FEET)
BETA-----CHANGE IN TROUGH WIDTH PER INCREMENTAL LENGTH
ZZERO-----FIN SURFACE LENGTH AREA WITH FIN PER UNIT LENGTH TO
AFOVAS-----RATIO OF SURFACE AREA WITHOUT FIN PER UNIT LENGTH
BOA-----SURFACE AREA WITHOUT FIN HEIGHT TO BASE OF FIN(COTANGENT ALFA)
OMEGA-----RATIO OF FIN HEIGHT TO BASE OF FIN(COTANGENT ALFA)
SECLNG-----ANGULAR VELOCITY(RADIAN PER HOUR)
              IF NO EXTENDED SURFACE, THE LENGTH OF THE SECTION OF
              INTEREST(FEET)
  
```

FOLLOWING CYLINDRICAL ANALYSIS CONTROL PARAMETERS ARE ALSO

```

INITIALIZED
ITER-----NUMBER OF MASS FLOW ITERATIONS
NDEL-----IS SET EQUAL TO 1 AFTER TEMPERATURE CONVERGENCE
              IS REACHED AT ALL INCREMENTS. IT ALLOWS FILM PROFILE
              SOLUTION TO ACCOUNT FOR TEMPERATURE VARIATION AXIALLY
NDELFN-----IS SET EQUAL TO 1 AFTER FINAL FILM PROFILE OF A MASS
              FLOW ITERATION IS DETERMINED PRIOR TO MASS FLOW CON-
              VERGENCE TEST.
NSTOP-----IS SET EQUAL TO 1 TO STOP MASS FLOW ITERATION
  
```


C

```

CL=CL I / 12.000
RBASE=RBASE I / 12.000
BFIN=BFIN I / 12.000
FNWDTH=FNWDTH I / 12.000
R2=RBASE-BFIN/2.000
PI=3.1415926535897900
PHI=2.000*CANGL*PI/360.000
SPHI=DSIN(PHI)
CPHI=DCCS(PHI)
TPHI=DTAN(PHI)
DIV=DFLCAT(NDIV)
DELX=CL/DIV
CBASE=2.000*PI*RBASE
SECLNG=CBASE/360.000
REXIT=REBASE*CL*TPHI
CEXIT=2.000*PI*REXIT
THICK=THICK I / 12.000
ALFA=FANGL*2.000*PI/360.000
SALFA=DSIN(ALFA)
CALFA=DCCS(ALFA)
TALFA=DTAN(ALFA)
OMEGA=RPM*2.000*PI*60.000
ITER=1
NDEL=0
NDELFN=C
IF (BFIN.NE.0) GO TO 60
BETA=((CEXIT-CBASE)/360.000)/DIV
SURFAR=CBASE
AFQVAS=1.000
BOA=0.000
GO TO 90
IF (NCMREC.EC.0) GO TO 70
EZERO=FNWDTH+2.000*BFIN*TALFA
EPSO=ETCEO*EZERO
ZF IN=CBASE/(EZERO+EPSO)
SURFAR=ZF IN*(FNWDTH+(2.000*(BFIN/CALFA)+EPSO))
AFQVAS=(FNWDTH+2.000*(BFIN/CALFA)+EPSO)/(EZERO*(1.000+ETUEO))
BOA=BFIN/(EZERO/2.000)
GO TO 80
CONTINUE
EZERO=2.000*BFIN*TALFA
EPSO=ETCEO*EZERO
ZF IN=CBASE/(EZERO+EPSO)
SURFAR=ZF IN*(2.000*(BFIN/CALFA)+EPSO)
AFQVAS=(ETUEO*(1./SALFA))/(1.+ETUEO)
CONTINUE

```

60

70

80


```

90      BOA=BFIN/(EZERO/2.000)
91      EPSEX=(CEXIT-(ZFIN*EZERO))/ZFIN
92      BETA=(EPSEX-EP50)/DIV
93      ZZERO=BFIN/CALFA
94      CONTINUE

      TEMPERATURE ESTIMATES ALONG INTERNAL CONVECTIVE BOUNDARY AND
      AVERAGE TEMPERATURES

T-----TEMPERATURE AT A NODAL POINT(DEG F)
TSOLID-----AVERAGE TEMPERATURE OF SOLID SECTION (DEG F)

DO 100 IGT=NFNTIP,NENTRF
NP=ICOR(IGT,2)
T(NP)=TINTL
CONTINUE
NP=ICOR(NENTRF,1)
T(NP)=TINTL
CONTINUE
TSOLID=((TSAT+TINF)/2.000

****      BEGIN MAIN ITERATIVE LOOP      ****

QFNTOT-----TOTAL HEAT RATE INTO THE FIN SECTIONS (BTU/HR)
QFTOT-----TOTAL HEAT RATE INTO TROUGH SECTIONS (BTU/HR)
QBOTOT-----TOTAL HEAT RATE OUT FROM BOTTOM OF ALL SECTIONS (BTU/HR)
QSMTOT-----TOTAL HEAT RATE INTO SMOOTH SECTION (BTU/HR)-NO EXTEND-
              ED SURFACE
DMTOT-----TOTAL CCNDENSATE MASS FLOW RATE (LBM/HR)
R-----MINIMUM RADIUS FOR A GIVEN INCREMENTAL SECTION (FEET)
SLNGTH-----LENGTH OF SMOOTH SECTION (FEET)
EPS-----TROUGH WIDTH FOR A GIVEN INCREMENTAL SECTION (FEET)
DLNGTH-----DISTANCE FROM CONDENSER END (X=0) TO MIDPOINT OF
              INCREMENT.

NOTE: ALL CALCULATIONS ARE FOR THE MIDPOINT OF AN INCREMENT

QFNTOT=C.0DC
QFTOT=C.0DC
QBOTOT=0.000
QSMTOT=C.0DC
DMTOT=0.000
DMDOT=0.000
DO 400 NI=1,NDIV
IF (NI.GT.1) GO TO 120

```



```

120 R(NI)=R2+DELX*SPHI/2.0D0
    EPS(NI)=EPSC+BETA/2.0D0
    SLNGTH(NI)=SECLNG+BETA/2.0D0
    DLNGTH(NI)=DELX/2.0D0
    GO TO 120
    CONTINUE
130 NI=NI-1
    R(NI)=R(NI1)+DELX*SPHI
    EPS(NI)=EPS(NI1)+BETA
    SLNGTH(NI)=SLNGTH(NI1)+BETA
    DLNGTH(NI)=DLNGTH(NI1)+DELX
    CONTINUE

***** TEMPERATURE DISTRIBUTION ALONG FIN *****
      NODAL POINT COORDINATES

NOTE: SYMMETRY BOUNDARY IS VERTICAL FROM BOTTOM EDGE OF SECTION
      (I.E. CUTSIDE WALL OF CONDENSER) TO APEX OF FIN BISECTING
      FIN INTO TWO EQUAL PARTS IF FINNED CONDENSER.
      IF SMOOTH CONDENSER, SYMMETRIC SECTION IS DEGREE ARC OF
      ROTATION OF CIRCUMFERENCE, I.E. 360 EQUAL SECTIONS.

X-----COORDINATE AXIS PERPENDICULAR TO SYMMETRY BOUNDARY
      ALONG BOTTOM EDGE OF SECTION
      (X=0 AT SYMMETRY BOUNDARY)
Y-----COORDINATE AXIS ALONG SYMMETRY BOUNDARY(Y=0 AT BOTTOM
      EDGE OF SECTION)
Z-----COORDINATE AXIS ALONG FIN SURFACE MEASURED FROM APEX
      OF FIN(Z=0 AT APEX)
ZB-----DISTANCE FROM APEX TO LOWER NODAL POINT OF MIDDLE
      ELEMENT ALONG CONVECTIVE BOUNDARY OF FIN

      DETERMINE X AND Y COORDINATES OF NODAL POINTS(USED FOR TEMPER-
      ATURE DISTRIBUTION DETERMINATION)
      CALL CCCR0

      ESTABLISH CONVERGENCE COUNTER
      IM=1
      INP=1
      Z(1)=0.0D0
      DO 140 IZEL=NFNTIP,NBSFIN
        NA=ICOR(IZEL,1)
        NB=ICOR(IZEL,2)
        XE=X(NA)-X(NB)
        YE=Y(NA)-Y(NB)
        ELZ=DSQRT(XE**2+YE**2)

```



```

140      Z(INP+1)=Z(INP)+ELZ
      I2=INP+1
      INP=INP+1
      CONTINUE
      XZB=X(ICOR(NFINM,1))-X(ICOR(NFNTIP,2))
      YZB=Y(ICOR(NFINM,1))-Y(ICOR(NFNTIP,2))
      ZB=DSQRT(XZB**2+YZB**2)
      ZC=ZZERC

C      PARABOLIC TEMPERATURE DISTRIBUTION ALONG FIN CONVECTIVE BOUN-
C      DARY USING LAGRANGE INTERPOLATION
C
C      TP1-----TEMPERATURE AT APEX
C      TP2-----TEMPERATURE AT LOWER NODAL POINT OF MIDDLE FIN CONVE-
C      CTIVE ELEMENT
C      TP3-----TEMPERATURE AT LOWER NODAL POINT OF ELEMENT AT BASE
C      OF FIN
C      TC-----SUM OF NODAL POINT TEMPERATURES FOR I NODAL POINTS
C      TFILM-----AVERAGE FILM TEMPERATURE OF CONDENSATE FILM ON FIN
C
150      CONTINUE
      IF (BFIN.EQ.0.000) GO TO 160
      TP1=T(ICOR(NFNTIP,2))
      TP2=T(ICOR(NFINM,1))
      TP3=T(ICOR(NBSFIN,1))
      AP1=TP1/(ZC*ZB)
      AP2=TP2/(ZB*(ZB-ZC))
      AP3=TP3/(ZC*(ZC-ZB))
      BP1=-(ZB+ZC)*AP1
      BP2=-ZC*AP2
      BP3=-ZB*AP3
      AA1=AP1+AP2+AP3
      BB1=BP1+BP2+BP3

C      CONTINUE
      TC=0.000
      DO 170 NY=NFNTIP,NENTRF
      NZ=ICOR(NY,2)
      TC=TC+T(NZ)
      CONTINUE
      RNY=DFLLCAT(NENTRF+1)
      NTRF1=ICOR(NENTRF,1)
      TFILM=(TC+T(NTRF1)+RNY*TSAT)/(2.000*RNY)

170      CONTINUE

C      ***** SOLID AND FLUID PROPERTIES *****
C      HFG-----LATENT HEAT OF VAPORIZATION (BTU/LBM)
C      RHO-----DENSITY OF FLUID (LBM/FT3)

```



```

UF----- DYNAMIC VISCOSITY OF LIQUID (LBM/FT-HR)
CF----- THERMAL CONDUCTIVITY OF CONDENSATE FILM (BTU/HK-FT-DEGF)
CW----- THERMAL CONDUCTIVITY OF CONDENSER WALL (BTU/HR-FT-DEGF)
CP----- SPECIFIC HEAT OF FLUID (BTU/LBM-DEG F)
RHOV----- DENSITY OF VAPOR (LBM/FT3)
UVAP----- DYNAMIC VISCOSITY OF VAPOR (LBM/FT-HR)

```

WATER PRCPERTIES

```

IF (IFLUID.EQ.1) GO TO 180
HFG=1093.88CO-0.5703D0*TSAT+0.00012819D0*(TSAT**2)-0.0000008824D0*
1(TSAT**3)
RHOQ(NI)=62.774D0-0.00255698D0*TFILM-0.000053572D0*TFILM**2
CF(NI)=0.3034D0+0.000738927D0*TFILM-0.0000147321D0*TFILM**2
UF(NI)=(0.001397D0-C.000014669D0*TFILM+0.0000003631253D0*TFILM**2-
1.0000000000576569D0*TFILM**3)*3600.0D0
CP(NI)=-0.000000000007D0*TFILM**3+0.0000014764D0*TFILM**2-0.000276
88D0*TFILM+1.01C9117D0
UVAP=(-0.0186C745C9D0+.000078977468D0*TSAT-1.5480676E-07*TSAT**2+4.3
820809E-10*TSAT**3)
RHOV=(-0.0118430827DC+.000335368496*TSAT-3.08706926E-06*TSAT**2+1.2
8265446E-08*TSAT**3)

```

FRECQ PRCPERTIES

```

IF (IFLUID.EQ.0) GO TO 190
HFG=69.5459-0.0156011*TSAT-0.000455294*(TSAT**2)+0.00000104144*(TS
1AT**3)
RHOQ(NI)=102.055-0.025364*TFILM-0.000502649*(TFILM**2)+0.000001354
107*(TFILM**3)
CF(NI)=0.871592253-0.000795216575*TFILM+6.5849702E-06*TFILM**2-1.85
886027E-08*TFILM**3
UF(NI)=(8.449682747E-04-7.85856781E-06*TFILM+4.2075531E-08*TFILM**
12-9.7346865E-11*TFILM**3)*3600.0D0
RHOV=-0.08682C129D0-.00159523356D0*TSAT+4.52222798E-05*TSAT**2-1.681
81776E-08*TSAT**3
UVAP=(-0.02264997421D0+.0000199556161*TSAT+.8031152E-07*TSAT**2-7.53
8704E-10*TSAT**3)
CP(NI)=-0.2106709091+.00016205808*TFILM+3.1628785E-07*TFILM**2-8.838
8385E-10*TFILM**3
CONTINUE

```

CALCULATE THERMAL CONDUCTIVITY OF WALL MATERIAL

```

IF (IFIN.EQ.1) GO TO 200
FOR CCPPER WALL MATERIAL

```

```

CW(NI)=231.7772CO-0.02222D0*TSOLID

```



```

200 IF (IFIN.EQ.0) GO TO 210
C
C FOR STAINLESS STEEL WALL MATERIAL
C
210 CW(NI)=6.776+0.00265*TSOLID
C CONTINUE
C
C ***** INITIAL FILM THICKNESS *****
C
C IF (PHI.EQ.0.0DO.AND.NI.NE.1) GO TO 220
C CALL DLSTAR(DLNGTH,RHOV,UWAP,CPHI,OMDOT,TAVERG,NDEL,NDELFN,ITER,IT
&ERMX,NSTOP,NONCE,TAVG,TI)
C IF (NSTCP.EQ.1) GO TO 410
C CONTINUE
C IF (NDEL.EQ.1) NDELFN=1
C
220
C
C ***** CALCULATE HEAT TRANSFER COEFFICIENTS OF *****
C CCNVECTIVE BOUNDARY ELEMENT SURFACES
C
C CALL HTCOEF(AA1,BB1,CPHI)
C
C ***** ENTRY INTO FINITE ELEMENT SOLUTION *****
C
C CALL FORMAF(A,F,NBAN)
C CALL BANDEC (A,F,NSNP,NBAN,1)
C
C ***** TEMPERATURE DISTRIBUTION *****
C T1-----TEMPERATURE AT FIN TIP (IF NO EXTENDED SURFACE, THE
SECTION-----TEMPERATURE AT THE INTERNAL LEFT CORNER OF THE UNIT
TBR-----TEMPERATURE AT EXTERNAL RIGHT CORNER OF SECTION
TBL-----TEMPERATURE AT EXTERNAL LEFT CORNER OF SECTION
TBM-----TEMPERATURE ON EXTENDED SURFACE DIRECTLY BELOW BASE
OF FIN, IF NO EXTENDED SURFACE, TEMPERATURE IN MIDDLE
OF EXTERNAL SURFACE
TBSFIN-----TEMPERATURE AT THE BASE OF THE FIN, IF NO EXTENDED
SURFACE, TEMPERATURE IN MIDDLE OF INTERNAL SURFACE
TTROF-----TEMPERATURE AT THE END OF THE TROUGH, IF NO EXTENDED
SURFACE, TEMPERATURE AT INTERNAL RIGHT CORNER OF
SECTION
C
C DO 230 AT=1,NSNP
C T(NT)=F(NT,1)
C CONTINUE
C T1(NI)=T1(CCR(NFNT,IP,2))
C TBR(NI)=T1(COR(NEXT,2))
230

```



```

C
C
C
C
      TBM(NI)=T(ICOR(NEXTM,2))
      TBL(NI)=T(ICOR(NEXTL,1))
      TTRF(NI)=T(ICOR(NENTRF,1))
      TBSFIN(NI)=T(ICCR(NBSFIN,1))

      DETERMINE NEW VALUE OF TSOLID

      SUMTMP=C*ODC
      DO 240 NS=1,NSNP
      SUMTMP=SUMTMP+T(NS)
      CONTINUE
      PN=DFLCAT(NSNP)
      TSOLID=SUMTMP/PN

C
C
C ***** CHECK CONVERGENCE OF TEMPERATURES *****
      IF (IM.EQ.1) GO TO 260
      DO 250 I=1,NSNP
      TJ(I)=T(I)
      CONTINUE
      GO TO 280
      CONTINUE
      DO 270 I=1,NSNP
      TI(I)=T(I)
      CONTINUE
      IM=2
      GO TO 150
      CONTINUE
      DO 290 I=1,NSNP
      DIFF(I)=ABS(TJ(I)-TI(I))/TJ(I)
      IF (DIFF(I).GT.CRIT) GO TO 300
      CONTINUE
      GO TO 320
      CONTINUE
      DO 310 I=1,NSNP
      TI(I)=TJ(I)
      CONTINUE
      GO TO 150
      CONTINUE

C
C
C
      DEFINE THE AVERAGE WALL TEMPERATURE OF AN INCREMENT
      IF (PHI.NE.0.000) GO TO 350
      TAVG(NI)=T(ICOR(NENTRF,1))
      IF (NI.NE.ND IV) GO TO 340
      TAVSUM=C*ODC
      DO 330 NR=1,ND IV
      TAVSUM=TAVSUM+TAVG(NR)

```



```

330 CONTINUE
340 TAVERG=TAVSUM/DFLOAT(NDIV)
350 CONTINUE
    IF (PHI.NE.0.000)NDEL=1
C ***** PERFORM HEAT TRANSFER CALCULATIONS *****
C CALL HTCALC
C *****
C ***** CALCULATE INCREMENTAL AND TOTAL MASS FLOW RATES *****
C DMDOT=2.000*QBI*DELX/HFG *****
C AMTOT=DMTCT+DMDCT *****
C DMTOT(NI)=ZFIN*DMTOT *****
C IF (NI.EC.NDIV)DMTOT=DMTOT*ZFIN *****
C GO TO 370 *****
C CONTINUE
360 DMDOT=QBI*DELX/HFG
    DMTCT=DMTCT+DMDCT
    AMTOT(NI)=ZFIN*DMTOT
    IF (NI.EC.NDIV)DMTOT=DMTOT*360.000
    CONTINUE
370 ***** DETERMINE NEXT INCREMENTAL TROUGH THICKNESS(DELTA STAR) *****
C NOTE: DLSTAR CALLED HERE TO DETERMINE FILM THICKNESS FOR NEXT IN-
C CREMENT IN TAPERED ANALYSIS. IN CYLINDRICAL ANALYSIS,DLSTAR
C CALLED HERE FOR MASS FLOW CONVERGENCE TEST.
C IF (PHI.EQ.C.000.AND.NI.NE.NDIV.OR.NDEL.NE.1)GO TO 380
C CALL DLSTAR(DLNGTH,RHUV,UVAP,CPHI,DMDOT,TAVERG,NDEL,NDELFN,ITER,IT
C &ERMN,NSTCIP,NONCE,TAVG,II)
C CONTINUE
380 STORE CCCORDINATES AND TEMPERATURES OF EACH INCREMENT
    DO 390 I=1,NSNP
    XPLT(NI,I)=X(I)
    YPLT(NI,I)=Y(I)
    TPLT(NI,I)=T(I)
    CONTINUE
390 CONTINUE
400

```



```

470 DO 470 NR=1,NDIV
C WRITE(6,950)NR,CINCSM(NR),QBINC(NR),QX(NR)
C CONTINUE
C
C      OUTPUT HEAT RATE INTO FIN, HEAT RATE INTO TROUGH, AND HEAT
C      RATE OUT BOTTOM AND TOTAL MASS FLOW RATE FOR A GIVEN SET
C      OF INPUT CONDITIONS
C      IF (BFIN.EQ.0.000) GO TO 490
C
C      IF (PHI.EQ.0.000) GO TO 480
C      WRITE(6,970) QFNTOT,QTFTOT,QBTOT,DMTOT
C      GO TO 510
C      CONTINUE
C      WRITE(6,960)QFNTOT,QTFTOT,QBTOT,DMTCT,FLOMAS
C      GO TO 510
C      IF (PHI.EQ.0.000)GO TO 500
C      WRITE(6,980)QSMTOT,QBTCT,DMTOT
C      GO TO 510
C      CONTINUE
C      WRITE(6,990) QSMTOT,QBTOT,DMTOT,FLCMAS
C      CONTINUE
C
C      OUTPUT INCREMENTALLY VARYING PROPERTIES
C
C      WRITE(6,005)
C      DO 520 NR=1,NDIV
C      WRITE(6,015) NR,CF(NR),CW(NR),UF(NR),RHOF(NR)
C      CONTINUE
C
C      OUTPUT INCREMENTALLY VARYING PARAMETERS
C
C      IF (BFIN.EQ.0.000) GO TO 540
C      WRITE(6,025)
C      DO 530 NR=1,NDIV
C      WRITE(6,035) NR,DELSTR(NR),EPS(NR),R(NR),AMTOT(NR)
C      CONTINUE
C      GO TO 560
C      CONTINUE
C      WRITE(6,045)
C      DO 550 NR=1,NDIV
C      WRITE(6,055) NR,DELSTR(NR),SLNGTH(NR),R(NR),AMTOT(NR)
C      CONTINUE
C      CONTINUE
C
C      OUTPUT MAJOR TEMPERATURES FOR EACH INCREMENT
C      IF (BFIN.EQ.0.000)GO TO 570

```



```

570 WRITE (6,065)
570 GO TO 580
570 CONTINUE
580 WRITE (6,075)
580 CONTINUE
580 DO 590 NR=1,NDIV
580 WRITE (6,085) NR,TBL(NR),TBM(NR),TBR(NR),TL(NR),TBSFIN(NR),TTROF(N
580 &R)
580 CONTINUE
580
580 OUTPUT NCDAL POINT X AND Y COORDINATES AND FINAL TEMPER-
580 TURE FOR EACH NODAL POINTFOR INCREMENTS CF INTEREST
580
580 WRITE (6,095)
580 DO 620 I=NFLAG1,NFLAG2,NFLAG3
580 WRITE (6,105) I
580 WRITE (6,115)
580 DO 600 NP=1,NSNP
580 WRITE (6,125) NP,XPLT(I,NP),YPLT(I,NP),TPLT(I,NP)
580 CONTINUE
580
580 OUTPUT CONVECTIVE BOUNDARY ELEMENT LENGTH,HEAT TRANSFER
580 COEFFICIENT AND HEAT RATE PER UNIT LENGTH FOR INCREMENTS OF
580 INTEREST
580
580 WRITE (6,135)
580 DO 610 IQEL=1,NEXTLT
580 WRITE (6,900) IQEL,ELMNT(I,IQEL),HELMNT(I,IQEL),QELMNT(I,IQEL)
580 CONTINUE
580 CONTINUE
580 IF (IUNITS.EQ.0) GO TO 640
580 CONTINUE
580
580 UNITS CONVERT ALL DIMENSIONAL QUANTITIES,INPUT AND CALCULATED,
580 TO SI UNITS AND OUTPUTS THESE QUANTITIES
580
580 CALL SIUNIT(TL,TPLT,XPLT,YPLT,NFLAG1,NFLAG2,NFLAG3)
580 CONTINUE
580
580
580 STOP
580 ***** INPUT FORMAT *****
580 FORMAT (8I5)
580 FORMAT (2I5)
580 FORMAT (6G1C.5)

```



```

680 FORMAT (2G1C.5)
690 FORMAT (2G1C.5)
700 FORMAT (5G1C.5)
710 FORMAT (4I5)
720 FORMAT (4I5)
730 FORMAT (2I5,2G15.10)
740 C
750 C
760 C
770 C
780 C
790 C
800 C
810 C
820 C
830 C
840 C
850 C
860 C

```



```

&FIN BASE WIDTH =, 01X,G10.5,1X,*,RATIO CF FIN HEIGHT TO FIN BASE =
&, 1X,G10.5,1X,*,RATIO OF SURFACE AREA TO SMOOTH AREA =, 01X,G10.5
&,,,
FORMAT (1H, 1X, 29HLATENT HEAT OF VAPORIZATION =,G10.5,8H BTU/LBM, /
11X, 20HFCR SATURATION TEMP=,G10.5,9HDEGREES F,/)
FORMAT (1X,*,VAPOR VISCOSITY AND DENSITY USED IN DRAG SOLUTION TERMS
&, //1X,*,VAPOR VISCOSITY =, 2X,F15.10,2X,*,LBM/FT-HR,*,1X,*,VAPOR DEN
&SITY =, 4X,F15.10,2X,*,LBM/FT3,/)
FORMAT (1H, 15X, 36HHEAT RATE SUMMARY FOR A UNIT SECTION, //1X, 10HIN
1CREMENT (1X, 15X, 10HHEAT RATE, 5X, 10HHEAT RATE, 5X, 9HHEAT
2T RATE, //15X, 8HINTO FIN, 6X, 11HINTO TROUGH, 6X, 8HTOTAL IN, 6X, 10HOUT B
3TOTOM, //15X, 8H(BTU/HR), 7X, 8H(BTU/HR), 8X, 8H(BTU/HR), 6X, 8H(BTU/HR), //)
FORMAT (1X, 15, 10X, G10.5, 3(5X, G10.5) //)
FORMAT (1X,*,RESULTS BASED ON BALLBACK EQUATION FOR TROUGH THICKNESS
&, //)
FORMAT (1X,*,RESULTS BASED ON DANIELS EQUATION FOR TROUGH THICKNESS
&, //1X,*,SIMPLIFIED FORM-FUNCTION OF SHERWOOD NUMBER ONLY, //)
FORMAT (1X,*,RESULTS BASED ON DANIELS EQUATION FOR TROUGH THICKNESS
&, //1X,*,FULL FORM, INCLUDES DRAG TERMS, //)
FORMAT (1H, 15X, 36HHEAT RATE SUMMARY FOR A UNIT SECTION, //1X, 10HIN
1CREMENT (1X, 15X, 10HHEAT RATE, 5X, 10HHEAT RATE, 5X, 9HHEAT
21X, 12HINTO SECTION, 4X, 14HOUT OF SECTION, 5X,*,FOR ALL SECTIONS, //14
32X, 8H(BTU/HR), 7X, 8H(BTU/HR), 9X,*,IN INCREMENT, //)
FORMAT (1X, 15, 10X, G10.5, 5X, G10.5, 9X, G10.5)
FORMAT (1H, 1X, 25HTOTAL HEAT RATE INTO FIN, 9X, F10.3, 7H BTU/HR, //1X
1, 27HTOTAL HEAT RATE INTO TROUGH, 8X, F10.3, 7H BTU/HR, //1X, 32HTOTAL HE
2AT RATE OUT OF HEAT PIPE, 3X, F10.3, 7H BTU/HR, //1X,*,TOTAL MASS FLOW
3AT RATE BASED ON HEAT RATE DIVIDED BY HFCR, F10.3, 3X,*,LBM/HR, //1X,*,TO
4TAL MASS FLOW RATE AT OVERFALL, F10.3, 3X,*,LBM/HR, //)
FORMAT (1H, 1X, 25HTOTAL HEAT RATE INTO FIN, 9X, F10.3, 7H BTU/HR, //1X
1, 27HTOTAL HEAT RATE INTO TROUGH, 8X, F10.3, 7H BTU/HR, //1X, 32HTOTAL HE
2AT RATE OUT OF HEAT PIPE, 3X, F10.3, 7H BTU/HR, //1X,*,TOTAL MASS FL
3OW RATE, 3X, G10.5, 7H LBM/HR, //)
FORMAT (1H, 1X, 31HTOTAL HEAT RATE INTO HEAT PIPE, 3X, F10.3, 7H BTU/
1HR, //1X,*,TOTAL MASS FLOW RATE BASED ON HEAT RATE DIVIDED BY HFCR, F10.3
2/1X,*,TOTAL MASS FLOW RATE BASED ON HEAT RATE DIVIDED BY HFCR, F10.3
3, 3X,*,LBM/HR, //1X,*,TOTAL MASS FLOW RATE AT OVERFALL, F10.3, 3X,*,LBM/
4HR, //)
FORMAT (1H, 10X, 43HFLUID AND MATERIAL PROPERTIES PER INCREMENT, //1X
1, 9HINCREMENT, 1, 7X, 5HFLUID, 15X, 4HWALL, 15X, 5HVISCOSITY, 12X, 7HDENSITY, 1
25X, 12HCONDUCTIVITY, 7X, 12HCONDUCTIVITY, //14X, 15HBTU/HR-FT-DEG F, 4X, 1
35HBTU/HR-FT-DEG F, 08X, 9HLBM/FT-HR, 12X, 7HLBM/FT3//)
FORMAT (1X, 15, 4(10X, G10.5),*,PARAMETERS PER INCREMENT, //1X, 10HINCREMENT
FORMAT (1H, 10X, 32HVARIOUS PARAMETERS PER INCREMENT, //1X, 10HINCREMENT

```



```

1NT ,6X,8HDEL STAR, 10X, 12HTROUGH WIDTH, 8X, 14HMINIMUM RADIUS, 6X, 14HM
2ASS FLOW RATE, /18X, 4HFEET, 16X, 4HFEET, 16X, 4HFEET, 16X, 6HLBM/HR, /)
FORMAT (1X, 15, 4(10X, G10.5))
FORMAT (//, 10X, 32HVARIOUS PARAMETERS PER INCREMENT, /1X, 10HINCREME
1NT ,6X, 8HDEL STAR, 10X, 14HSECTION LENGTH, 6X, 14HMINIMUM RADIUS, 6X, 14
2HMASS FLOW RATE, /18X, 4HFEET, 14X, 4HFEET, 16X, 6HLBM/HR, /)
FORMAT (1X, 15, 4(10X, G10.5))
FORMAT (//, 17X, MAJOR TEMPERATURES PER UNIT SECTION FOR ALL INCREME
ENTS, /, 7X, EXTERNAL, 07X, EXTERNAL, 07X, INTERNAL, 07X, INTERNAL,
66X, INTERNAL, 7X, INTERNAL, /, 8X, LEFT, 08X, BELOW BASE, 08X, RIGHT,
8, 09X, FIN TIP, 07X, FIN BASE, 07X, TROUGH END, /, 7X, DEGREES F, 0
66X, DEGREES F, 06X, DEGREES F, 05X, DEGREES F, 06X
8, DEGREES F, /)
FORMAT (//, 17X, MAJOR TEMPERATURES PER UNIT SECTION FOR ALL INCREME
ENTS, /, 7X, EXTERNAL, 07X, EXTERNAL, 07X, INTERNAL, 07X, INTERNAL,
66X, INTERNAL, 7X, INTERNAL, /, 8X, LEFT, 08X, MIDDLE, 08X, RIGHT,
8, 09X, LEFT, 07X, MIDDLE, 07X, RIGHT, /, 7X, DEGREES F, 0
66X, DEGREES F, 06X, DEGREES F, 05X, DEGREES F, 06X
8, DEGREES F, /)
FORMAT (1X, 12, 4(5X, G10.5), 2(4X, G10.5))
FORMAT (1H0, //, 10X, 63HMODAL POINT COORDINATES AND TEMPERATURE AT
1A SPECIFIC HEAT INCREMENT, /, 10X, 58HHEAT TRANSFER COEFFICIENT, ELEMENT LE
2NGTH AND HEAT RATE AT, /, 10X, 55HCONVECTIVE BOUNDARY ELEMENTS AT TH
3IS SPECIFIC INCREMENT, /)
FORMAT (1H0, 10X, 17HINCREMENT NUMBER, 15, /)
FORMAT (10X, 39HMODAL POINT COORDINATES AND TEMPERATURE, /, 1X, 11HMOD
1AL POINT, 5X, 8HX-COORD, 09X, 7HY-COORD, 9X, 11HTEMPERATURE, /18X, 4HFEET
2, 13X, 4HFEET, 12X, 9HDEGREES F, /)
FORMAT (1X, 15, 4X, 3(07X, G10.5))
FORMAT (1H0, 10X, 38HCONVECTIVE BOUNDARY ELEMENT PARAMETERS, /01X, 11
1HELEMENT NR, 10X, 6HLENGTH, 10X, 13HHEAT TRANSFER, 04X, 25HHEAT RATE PE
2R UNIT LENGTH, /23X, 4HFEET, 12X, 11HCOEFFICIENT, 13X, 9HBTU/HR-FT, /39X,
312HBTU/HR-FT, /)
FORMAT (01X, 15, 3(12X, G10.5))
END

```

```

145
C
C
C
C
SUBROUTINE CORRESPONDENCE BETWEEN ELE-
MENTS AND NODAL POINTS AND DEFINES MAJOR FINITE ELEMENT
PARAMETERS

```

```

SUBROUTINE CORRES(NPRNT,NBAN,NEXTM,NFINM)
IMPLICIT REAL*8(A-H,O-Z)
COMMON /GLOB1/ BOA,BFINI,CANGL,CLI,ETOEO,FANGL,HINF,QBTOT,RBASEI,R
&PM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
COMMON /GLOB2/ AFOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
1ITDL,CW(100),DELSTR(100),DELX,DMQI,ELMNT(100,50),EPS(100),EZERO,F
2LOMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
3(100,50),QFANTOT,QINC(100),QINCSM(100),QSMTOT,QTFIOT,QTINC(100),QTO
4TAL(100),QX(100),R(100),RHOF(100),SALFA,SLNGTH(100),SPHI,SURFAR,TI

```


5100), TALFA, TBL(100), TBM(100), TBR(100), TBSFIN(100), TTRUF(100), THICK
6,UF(100),X(100),Y(100),Z(100),ZCGR(200,3),NEXTLT,NEXTRT,NBSF
7IN,NCMTRF,NDIV,NEL,NENTRF,NI,NPDIFF,NPORIG,NPFCNV(10),NPFSY
8M(10),NPSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
COMMON/ECT/FNWIDTH,NFNTIP

NOTE: ELEMENTS ARE NUMBERED IN THE FOLLOWING ORDER:

- 1) STARTING WITH APEX OF FIN
- 2) ALONG INSIDE CONVECTIVE BOUNDARY FROM TOP TO BASE OF FIN
AT TROUGH FROM BASE OF FIN TO END OF TROUGH
- 3) ALONG OUTSIDE CONVECTIVE BOUNDARY FROM RIGHT TO LEFT
- 4) ALL OTHERS FROM TOP TO BOTTOM, LEFT TO RIGHT

NCCAL POINTS OF AN ELEMENT ARE NUMBERED IN THE FOLLOWING ORDER:

- 1) NODAL POINTS ARE NUMBERED COUNTERCLOCKWISE
- 2) ELEMENTS WITH A CONVECTIVE BOUNDARY HAVE ELEMENT NODAL POINTS 1 AND 2 LOCATED ON THE CONVECTIVE BOUNDARY
- 3) ELEMENTS NOT ON A CONVECTIVE BOUNDARY HAVE ELEMENT NODAL POINT NR. 1 IS THE LOWER LEFT NODAL POINT OF THE ELEMENT

NEL-----NUMBER OF ELEMENTS
NSNP-----NUMBER OF SYSTEM NODAL POINTS
NBAN-----SYSTEM BANDWIDTH
NENTRF-----NUMBER OF THE ELEMENT OF A SECTION, WITH INTERNAL
CONVECTIVE BOUNDARY, LOCATED AT END OF TROUGH
IF THERE IS NO EXTENDED SURFACE, NENTRF IS THE RIGHT
HAND ELEMENT ON THE INTERNAL CONVECTIVE BOUNDARY
NBSFIN-----NUMBER OF THE ELEMENT OF A SECTION WITH INTERNAL
CONVECTION BOUNDARY LOCATED AT BASE OF FIN
IF NO EXTENDED SURFACE, NBSFIN IS THE MIDDLE ELEMENT
ON THE INTERNAL CONVECTIVE BOUNDARY
NEXTRT-----NUMBER OF THE ELEMENT WITH EXTERNAL CONVECTIVE
BOUNDARY LOCATED AT RIGHT BOTTOM CORNER OF SECTION
NEXTLT-----NUMBER OF THE ELEMENT WITH EXTERNAL CONVECTIVE
BOUNDARY LOCATED AT LEFT BOTTOM CORNER OF SECTION
NEXTF-----NEXTLT PLUS 1
NEXTM-----NUMBER OF INTERMEDIATE ELEMENT ON BOTTOM WITH SECOND
NODAL POINT LOCATED DIRECTLY BELOW FIRST NODAL POINT
OF ELEMENT AT BASE OF FIN (NBSFIN)
NF INM-----NUMBER OF THE INTERMEDIATE ELEMENT OF FIN ON THE IN-
TERNAL CONVECTIVE BOUNDARY WHOSE FIRST NODAL POINT
IS IN THE MIDDLE OR ADJACENT TO THE MIDDLE OF FIN
ALONG THE Z AXIS
NFNTIP-----NUMBER OF ELEMENT AT THE TIP OF THE FIN ON THE VERTI-
CAL CONVECTIVE SURFACE. NFNTIP WILL BE EQUAL TO 1 IF
ONLY A TRIANGULAR FIN
NRWFIN-----NUMBER OF ROWS IN THE FINSECTION

CC


```

NCMTRI-----NUMBER OF COLUMNS IN THE TRIANGULAR PORTION OF THE
FIN(MAY BE EQUAL TO 0 IF ONLY RECTANGULAR FIN)
NRWTRF-----NUMBER OF ROWS IN THE TROUGH SECTION
NCMTRF-----NUMBER OF COLUMNS IN THE RECTANGULAR PORTION OF THE
NCMREC-----NUMBER OF COLUMNS IN THE RECTANGULAR FIN)
NRFIN-----FIN(MAY BE EQUAL TO 0 IF ONLY A TRIANGULAR FIN)
NCFIN-----NUMBER OF ROWS IN THE FIN SECTION MINUS 1
FIN PLUS 1
NCOL-----TOTAL NUMBER OF COLUMNS IN THE FINITE ELEMENT MODEL
IF AN EXTENDED SURFACE EXISTS, NCOL IS EQUAL TO THE
SUM OF THE NUMBER OF COLUMNS IN THE FIN AND THE
TROUGH
NPDIFF-----NUMERICAL DIFFERENCE BETWEEN TWO ADJACENT VERTICAL
SYSTEM NODAL POINTS IN TROUGH SECTION
NPFSYM-----SYSTEM NODAL POINTS LOCATED ON SYMMETRY BOUNDARY
NPFCNV-----SYSTEM NODAL POINTS LOCATED ALONG FIN CONVECTIVE
BOUNDARY
NPORIG-----SYSTEM NODAL POINT LOCATED AT ORIGIN OF COORDINATE
SYSTEM
NPSMBS-----SYSTEM NODAL POINT LOCATED AT JUNCTION OF SYMMETRY
BOUNDARY AND LINE OF INTERSECTION OF BASE OF FIN
ICOR-----SYSTEM NODAL POINT CORRESPONDING TO ITH NODAL POINT
OF ELEMENT IEL

```

IDENTIFY MAJOR ELEMENTS NUMBERS ALONG CONVECTIVE BOUNDARY

```

NEL=2*NROWTRF*NCOL+2*NROWFIN*NCMREC+NCMTRI**2
NFNTIP=NCMREC+1
IF (NRWFIN.NE.0) GO TO 10
NBSFIN=NCOL/2
IF (DFLCAT(NCOL)/2.000.GT.DFLUAT(NCOL/2)) NBSFIN=NBSFIN+1
NENTRF=NCOL
GO TO 20
CONTINUE
NBSFIN=NCMREC+NROWFIN
NENTRF=NBSFIN+NCMTRF
CONTINUE
NEXTRI=NENTRF+1
NEXLTR=NEXTRI+NCOL-1
NFINM=NENTRF+(NBSFIN-NFNTIP)/2
NEXTM=NEXTRI+(NEXLTR-NEXTRI)/2

```

IDENTIFY OTHER MAJOR ELEMENTS THAT ARE USED AS COUNTERS

```

NRFIN=NROWFIN-1

```



```

NCFIN=NCMTRI-1
NCMI=NCMREC+1
IF (NRWFIN.NE.0) GO TO 30
NCMREC=NCOL
GO TO 40

IF (NCMREC.EQ.0) GO TO 90
CONTINUE

DO 50 I=1,NCMREC
  ICOR(I,1)=I+1
  ICOR(I,2)=I
  ICOR(I,3)=I+1+NCMREC
CONTINUE
IF (NRWFIN.EQ.0) GO TO 120

IF (NCMTRI.EQ.0) GO TO 70
NFINST=NCMREC+1
DO 60 I=NFINST,NBSFIN
  ICOR(I,2)=ICOR(I-1,1)
  ICOR(I,1)=ICOR(I,2)+I+1
  ICOR(I,3)=ICOR(I,1)-1
CONTINUE
GO TO 100
CONTINUE

NEXT=NCMREC+1
DO 80 I=NEXT,NBSFIN
  ICOR(I,2)=ICOR(I-1,1)
  ICOR(I,1)=ICOR(I,2)+NCMREC+1
  ICOR(I,3)=ICOR(I,1)-1
CONTINUE
GO TO 100
CONTINUE
J=1
JJ=3
DO 100 I=1,NRWFIN
  ICOR(I,1)=JJ
  ICOR(I,2)=J
  ICOR(I,3)=JJ-1
  J=JJ
  JJ=JJ+I+2
CONTINUE

NN=NBSFIN+1
JJ=ICCR(NBSFIN,1)+1
DO 110 I=NN,NENTRF
  ICOR(I,1)=JJ

```



```

110 ICOR(IJ,2)=JJ-1
110 ICOR(IJ,3)=ICOR(IJ,1)+NCOL
110 JJJ=JJJ+1
110 CONTINUE
120 CONTINUE
120 NJ=ACCL+1
120 IF (NRWFIN.EQ.0) GO TO 130
120 JJ=(NRWTRF+1)*NJ
120 GO TO 140
130 CONTINUE
130 JJ=JJJ+NRWTRF*NJ-1
140 CONTINUE
140 J=JJ-1
140 DO 150 IJ=NEXTRT,NEXTLT
140 ICOR(IJ,1)=J
140 ICOR(IJ,2)=JJ
140 ICOR(IJ,3)=JJ-NJ
140 JJ=JJ-1
140 J=J-1
140 CONTINUE
150 IF (NCMREC.EQ.0) GC TO 190
150 II=ICOR(1,3)
150 III=ICOR(1,1)
150 JJ=NEXTLT+1
150 JJJ=JJJ+NCMREC-1
150 IF (NCMTRI.EQ.0) JJJ=JJJ-1
150 DO 160 IJ=JJ,JJJ
150 ICOR(IJ,1)=II
150 ICOR(IJ,2)=II+1
150 ICOR(IJ,3)=III
150 II=II+1
150 III=III+1
150 CONTINUE
150 IJK=JJJ
150 IF (NRWFIN.EQ.0) GO TO 240
160 IF (NRWFIN.EQ.1) GO TO 210
160 NRREC=NRWFIN-1
160 IJK=JJJ
160 II=ICOR(1,3)
160 DO 180 I=1,NRREC
160 IJ=NCMREC+I

```



```

IF (NCMTRI.EQ.0) IJ=NCMREC
DO 170 J=1,IJ
  IJK=IJK+1
  ICOR(IJK,3)=II+J-1
  ICOR(IJK,2)=ICOR(IJK,3)+1
  ICOR(IJK,1)=ICOR(IJK,2)+1
  IF (NCMTRI.EQ.0.AND.CJ.EQ.NCMREC) GO TO 170
  IJK=IJK+1
  ICOR(IJK,1)=ICOR(IJK-1,1)
  ICOR(IJK,2)=ICOR(IJK,1)+1
  ICOR(IJK,3)=ICOR(IJK-1,2)
  CONTINUE
  II=II+IJ+1
  CONTINUE
  JJ=IJK
  GO TO 210
170
180
C
190
CONTINUE
NN=NRWFIN-1
II=4
JJ=NEXTLT
DO 210 I=1,NN
  DO 200 J=1,I
    JJ=JJ+1
    ICOR(JJ,1)=II+J-1
    ICOR(JJ,2)=ICOR(JJ,1)-1
    ICOR(JJ,3)=ICOR(JJ,2)-1
    JJ=JJ+1
    ICOR(JJ,1)=II+J-1
    ICOR(JJ,2)=ICOR(JJ,1)+1
    ICOR(JJ,3)=ICOR(JJ-1,2)
    CONTINUE
    II=II+I+2
    CONTINUE
  IF (NCMREC.NE.0) GO TO 220
  II=NRWFIN*(NRWFIN+1)/2+(NCOL+2)
  GO TO 240
  IF (NCMTRI.NE.0) GO TO 230
  II=NRWFIN*(NCMREC+1)+(NCOL+2)
  GO TO 240
  CONTINUE
  II=(NRWFIN*NCMREC)+NRWFIN*(NRWFIN+1)/2+(NCOL+2)
  CONTINUE
  IF (NRWFIN.NE.0) GO TO 250
  II=ICOR(1,3)
  JJ=NEXTLT
  NCMREC=0
200
210
C
220
230
240

```


[illegible]


```

330 IF (NCMTRI.EC.O) GO TO 330
    NPFCNV(KKI)=NPFCNV(KKI-1)+NCMREC+KKI+1
    GO TO 340
340 CONTINUE
350 NPFCNV(KKI)=NPFCNV(KKI-1)+NCMREC+1
    CONTINUE
360 CONTINUE
    NPORIG=ICOR(NEXTLT,1)
    NPDIFF=(ICOR(NEXTLT,2)-ICOR(NENTRF,1))/NRWTRF
    NSNP=1
    NBAN=1
    IF (NPRNT.EQ.1) WRITE (6,390)
    PRINT ELEMENTS AND CORRESPONDING NODAL PCINTS
    DO 370 IK=1,NEL
    IF (NPRNT.EQ.1) WRITE (6,400) IK, ICOR(IK,1), ICOR(IK,2), ICOR(IK,3)
    NSNP=AMAX0(ICOR(IK,1), ICOR(IK,2), ICOR(IK,3), NSNP)
    II=IABS(ICOR(IK,1)-ICOR(IK,2))
    JJ=IABS(ICOR(IK,2)-ICOR(IK,3))
    KK=IABS(ICOR(IK,1)-ICOR(IK,3))
    NBAN=MAX0(NBAN, II, JJ, KK)
    CONTINUE
    NBAN=NBAN+1
    PRINT TOTAL NUMBER OF SYSTEM NODAL PCINTS
    IF (NPRNT.NE.1) GO TO 380
    WRITE (6,410) NSNP, NBAN
    PRINT MAJOR ELEMENT NUMBERS
    WRITE (6,420) NBSFIN, NENTRF, NEXTLT, NEXTLT
    PRINT NODAL POINTS FOR FOLLOWING LOCATIONS:
    1) BOTTOM RIGHT CORNER
    2) BOTTOM MIDDLE
    3) BOTTOM LEFT CORNER
    4) END OF TROUGH
    5) BASE OF FIN
    WRITE (6,430) ICOR(NEXTLT,2), ICOR(NEXTLT,1), ICOR(NEN

```



```

1TRF,1),ICOR(NBSFIN,1)
CONTINUE

380
C
C
390
RETURN (4X,11HELEMENT NR.,4X,17HNODAL PCINT NR. i,4X,17HNODAL POIN
FORMAT 2,4X,17HNODAL POINT NR. 3//)
400
FORMAT(6X,1,3(18X,13),/)
410
FORMAT(1X,30HNUMBER OF SYSTEM NODAL POINTS=,I3,/,1X,17HSYSTEM BAN
1DWIDTH=,13X,13,//)
420
FORMAT(05X,21HMAJOR ELEMENT NUMBERS,/,1X,12HNOMENCLATURE,6X,12HELE
MENT NR.,/,1X,9HFIN BASE ,10X,15,/,1X,14HEND OF TROUGH ,5X,15,/,1X,1
22HBOTTOM RIGHT,7X,15,/,1X,11HBOTTOM LEFT,8X,15,//)
430
FORMAT(05X,39HMAJOR TEMPERATURE LCCATICN NODAL POINT ,/1X,8HLOCAT
ION,10X,11HNODAL POINT,/,1X,12HBOTTOM RIGHT,7X,15,/,1X,17HBOTTOM BEL
20W BASE,2X,15,/,1X,11HBOTTOM LEFT,8X,15,/,1X,13HEND OF TROUGH,6X,15,
3/1X,10HFIN BASE ,5X,15,//)
END

SUBROUTINE CCORD ESTABLISHES X AND Y COORDINATES FOR NODAL
POINTS

SUBROUTINE CCORD
IMPLICIT REAL*8(A-H,O-Z)
COMMON /GLOBE1/ BOA,BFINI,CANGL,CLL1,ETOEO,FANGL,HINF,QBTOT,RBASEI,R
&PM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
COMMON /GLOBE2/AFDVAS,AMDT(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
1ITDL,CW(100),DELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
2LOMAS,H(200),HELMT(100,50),HOF(100),SALFA,SLNGT,H(100),SPHI,SURFAR,T(
3(100,50),QFNUT,QINC(100),QINCSM(100),QSMTOT,QTFTOT,QTINC(100),QTO
4TAL(100),QX(100),R(100),RHO(100),TBR(100),TBSFIN(100),TTRUF(100),THICK
5100),TALFA,TBL(100),TBM(100),Z(100),ZZERO,ICOR(200,3),NEXTLT,NEXTRI,NBSF
6UF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NEXTLT,NEXTRI,NBSF
7IN,NCMTRI,NCMTRF,NDIV,NEL,NENIRF,NINPDIFF,NPORG,NPFCNV(10),NPFSY
8M(10),NPFSBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
COMMON/RECT/FNWDTH,NFNTIP

IF (NRWFIN.NE.0) DELH=BFIN/DFLOAT(NRWFIN)
X(1)=0.C00
Y(1)=THICK+BFIN
IF (NRWFIN.EQ.0) GO TO 180

IF (NCMREC.EQ.0) GO TO 60
DELTAX=FNWDTH/(DFLOAT(NCMREC)*2.000)
NCREC1=NCMREC+1
DO 10 I=2,NCREC1
X(I)=X(I-1)+DELTAX
Y(I)=Y(I-1)
CONTINUE
10
C
C
C

```



```

190      DO 200 I=1,NROW1
      DO 190 J=2,NCOL1
      X(J+JJ)=X(J+JJ-1)+DEL TAX
      Y(J+JJ)=Y(J+JJ-1)
      CONTINUE
      JJ=1+NCCL1
      IF (I.EC.NRCW1) GO TO 200
      X(JJ+1)=X(1)
      Y(JJ+1)=Y(JJ)-DELY
      IF (I.EQ.NROW) Y(JJ+1)=0.000
      CONTINUE
      RETURN
      END
      SUBROUTINE DLSTAR DETERMINES THE FILM THICKNESS ON THE SMOOTH
      SURFACE OF THE HEAT PIPE
      C
      C
      C
      SUBROUTINE DLSTAR(DLNGTH,RHOV,UVAP,CPHI,DMDCT,TAVERG,NDEL,NDELFN,I
      &TER,IITRMX,ASTOP,NONCE,TAVG,T1)
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION XCOF(5),ROOTR(4),ROOTI(4),COF(5),FEGPRM(100),REYN(100)
      DIMENSION VEL(100),TAVG(100),T1(100),DLNGTH(100)
      COMMON /GLOB1/ BOA,BFINI,CANGL,CLI,EIOEO,FANGL,HINF,QBTOT,RBASEI,R
      &PM,R21,THICK1,TINF,TINTL,TSAT,ZFIN
      COMMON /GLOB2/ AFOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CKIT,CR
      1ITDL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
      2LOMAS,CH(200),HELMNT(100,50),HFG,OMEGA,PFI,PI,QBI,QBINC(100),QUELMNT
      3(100,50),QENTOT,QINC(100),QINCSM(100),QSMTOI,QTFUT,QTINC(100),QTQ
      4TAL(100),QX(100),R(100),RHO(100),SALFA,SLNGTH(100),SPHI,SURFAR,TC(
      5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TIRUF(100),THICK
      6UF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NE,NPFCNV(10),NPFSY
      7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NI,NPCIFF,NPORIG,NXTR,NPFSY
      8M(10),NPSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,C,NCOL,NSOLVE
      COMMON/DEL1/RELAX,DELMAX,IITRPT
      HFGPRM-----CORRECTED LATENT HEAT OF VAPORIZATION
      SHER-----SHERWOOD NUMBER
      C
      C
      C
      HFGPRM(NI)=HFG+0.3500*CP(NI)*(TSAT-T(ICOR(NENTRF,2)))
      SHER=(RHC(NI))*2*(OMEGA**2*R(NI))-(32.174*3600.0**2))*SPHI*
      *HFGPRM(NI)*DLNGTH(NI)**3/(4.0000*UF(NI)*CF(NI)*(TSAT-T(ICOR(NENTRF
      *2)))
      *IF (PHI.EQ.0.000) GO TO 200
      DELSTR-----CONDENSATE THICKNESS IN TROUGH
      C
      C
      C
      THIS PORTION OF DLSTAR DETERMINES TROUGH FILM THICKNESS FOR TRUNC-
      ATED HEAT PIPE

```



```

10 IF (NDEL.EQ.1) GO TO 10
10 C IF (NI.NE.1) GO TO 440
10 C DELSTR(1)=1.107*(DABS((TSAT-TINF)*CF(NI)/(UF(NI)*HFG))**.25)*{(DABS
10 C 1(UF(NI)/(RHC(NI)*OMEGA))**.5)
10 C GO TO 440
10 C CONTINUE

20 DELTA STAR IS SMALLEST POSITIVE REAL ROOT OF FOURTH DEGREE POLY-
20 C NOMIAL EQUATION IN THE CASE OF THE FINNED TAPERED HEAT PIPE
20 C IF NSOLVE IS EQUAL TO 1, THIS ROOT WILL BE FOUND

20 C A POLYNOMIAL ROOTFINDER ROUTINE IS USED TO FIND ROOTS
20 C XCOF-----COEFFICIENT OF POLYNOMIAL
20 C CL-----CONSTANT OF TWO COEFFICIENTS
20 C IF (BFIN.NE.C.OOO) GO TO 20
20 C IF (NSOLVE.NE.1) GO TO 50
20 C CONTINUE
20 C CL=RHOF(NI)**2*CMEGA**2*R(NI)/(3.OOO*UF(NI))*SPHI
20 C XCOF(1)=-DMTOT
20 C XCOF(2)=0.OOO
20 C XCOF(3)=0.OOO
20 C XCOF(4)=CL*EPS(NI)
20 C XCOF(5)=CL*TALFA
20 C M=4
20 C IF (CALFA.EQ.1.OOO) M=3
20 C CALL DPCLRT (XCOF, COF, M, ROOTR, ROOT1, IER)
20 C IF (NI.GT.1) GO TO 30
20 C WRITE (6,150)
20 C CONTINUE
20 C WRITE (6,160) (ROOTR(I), I=1,4)
20 C IF (ROOTR(1).GT.0.OOO) GO TO 40
20 C IF (ROOTR(2).GT.0.OOO) GO TO 50
20 C IF (ROOTR(3).GT.0.OOO) GO TO 60
20 C IF (ROOTR(4).GT.0.OOO) GO TO 70

30 C DEFINE NEW TROUGH CONDENSATE THICKNESS

30 C DELSTR(NI+1)=ROOTR(1)
30 C GO TO 80
30 C DELSTR(NI+1)=ROOTR(2)
30 C GO TO 80
30 C DELSTR(NI+1)=ROOTR(3)
30 C GO TO 80
30 C DELSTR(NI+1)=ROOTR(4)
30 C CONTINUE
30 C NDEL=0
30 C GO TO 440

```



```

C
C
C
C 90
IF NSOLVE EQUALS 2, BALLBACKS EQUATION IS USED TO DETERMINE DELTA
STAR FOR EACH INCREMENT IN THE SMOOTH TAPERED HEAT PIPE ANALYSIS
IF(NSOLVE.NE.2)GO TO 100
A1=(RBASEI/12.0D0)/((RBASEI/12.0D0)+DLNGTH(NI)*SPHI)
A2=CF(NI)*UF(NI)*((TSAT-T(3))/(RHOF(NI)**2*OMEGA**2*SPHI**2*HFG))
A22=(3.0D0/2.0D0)*A2
C=8.0D0/3.0D0
DELSTR(NI+1)=DABS(A22*(1-DABS(A1)**C))**.25
NDEL=0
GO TO 44C

C
C
C
C 100
IF NSOLVE EQUALS 3, THE DANIELS AND AL-JUMAILY SOLUTION NEGLECTING
DRAG IS USED TO SOLVE FOR DELTA STAR AT EACH INCREMENT OF THE
SMOOTH TAPERED HEAT PIPE
IF (NSOLVE.NE.3) GO TO 110
DELST=DABS(1.0D0/SHER)**.25
DELSTR(NI+1)=DELST*DLNGTH(NI)
NDEL=0
GO TO 440

C
C
C
C 110
IF NSOLVE EQUALS 4, THE DANIELS AND AL-JUMAILY SOLUTION INCLUDING
DRAG DUE TO VAPOR FRICTION TO DETERMINE DELTA STAR FOR THE SMOOTH
TAPER HEAT PIPE
CONTINUE
VEL-----VELOCITY(FT/HR)
VELO-----VELOCITY(FT/SEC)
REYN-----REYNOLDS NUMBER
FRIC-----FRICTION FACTOR
TAUV-----SHEAR STRESS VAPOR-LIQUID INTERFACE
DRX-----DRAG NUMBER
REVX-----TWO PHASE REYNOLDS NUMBER
VEL(NI)=(36C.0D0*DMTGT/(RHOF*PI*R(NI)**2))
VELO=VEL(NI)/3600.0D0
REYN(NI)=(DMTGT*360.0D0/UVAP)*(4.0D0/(PI*2.0D0*R(NI)))
IF (REYN(NI).GT.2000.0) GO TO 120
FRIC=16.0D0C/REYN(NI)
GO TO 130
CONTINUE
FRIC=.0791/(DABS(REYN(NI))**.25)
CONTINUE
TAUV=.05C*FRIC*RHOF*VEL(NI)**2
DRX=RHOF(NI)*TAUV*HFGPRM(NI)*DLNGTH(NI)**2*CPHI/(UF(NI)*CF(NI)*
&(TSAT-T(3)))

```



```

REXV=RHCF(NI)*VEL(NI)*DLNGTH(NI)*CPHI/UF(NI)
XCOF(1)=-1.000
XCOF(2)=0.000
XCOF(3)=((-1.000/4.000)*REXV)
XCOF(4)=((-1.000/3.000)*DRX)
XCOF(5)=SHEF
M=4

```

```

C      CALL DPCLRT (XCCF,CQF,M,ROOTR,ROOTI,IER)
C      IF (NI.GT.1) GO TO 140
C      WRITE (6,45C)

```

```

C      CONTINUE
C      WRITE (6,46C) (ROOTR(I),I=1,4)
C      IF (ROOTR(1).GT.0.000) GO TO 150
C      IF (ROOTR(2).GT.0.000) GO TO 160
C      IF (ROOTR(3).GT.0.000) GO TO 170
C      IF (ROOTR(4).GT.0.000) GO TO 180

```

```

C      DEFINE NEW TROUGH CONDENSATE THICKNESS

```

```

C      DELSTR(NI+1)=ROOTR(1)
C      GO TO 150
C      DELSTR(NI+1)=ROOTR(2)
C      GO TO 150
C      DELSTR(NI+1)=ROOTR(3)
C      GO TO 150
C      DELSTR(NI+1)=ROOTR(4)
C      CONTINUE
C      IF (BFIN.EQ.0.000.AND.ROOTR(4).GT.ROOTR(3))DELSTR(NI+1)=ROOTR(4)
C      DELSTR(NI+1)=DELSTR(NI+1)*DLNGTH(NI)
C      NDEL=0
C      GO TO 440

```

```

C      CALCULATE FILM THICKNESS FOR CYLINDRICAL HEAT PIPE

```

```

C      CONTINUE

```

```

C      SUBROUTINE DELCRV SOLVES FOR FILM PROFILE OF CYLINDRICAL HEAT
C      PIPE. DELCRV WILL DETERMINE THE FILM PROFILE ALONG THE HEAT PIPE
C      AND THE DERIVATIVE AT THE OVERFALL. THE PROGRAM, AS WRITTEN, WILL
C      DETERMINE A NEW PROFILE FOR EACH ITERATION WHEN NI EQUALS 1 UN-
C      TIL TEMPERATURE CONVERGENCE IS REACHED AT NI EQUALS 1. THIS PRO-
C      FILE WILL BE MAINTAINED FOR NI EQUALS 2-NDIV. A NEW PROFILE WILL
C      THEN BE DETERMINED USING THE AVERAGED TEMPERATURE AT EACH ITERA-
C      TION. THIS PROFILE WILL BE MAINTAINED FOR NI EQUALS 1-NDIV UNTIL
C      MASS FLOW RATES ARE COMPARED. IF CONVERGENCE IS NOT MET, THE PRO-
C      CESS WILL START ANEW.

```



```

C
C
C
C
C
C
NOTE:NDEL IS A CONTROL NUMBER, IT IS SET EQUAL TO 1 AFTER NI HAS
ITERATED TO NDIV.WHEN NDEL IS EQUAL TO ONE, THE AVERAGE TEMPERA-
TURE FROM THE FIRST COMPLETED ITERATION IS USED TO DETERMINE THE
CONSTANTS IN DELCRV
NOTE:NDELFN IS A CONTROL NUMBER, IT IS SET EQUAL TO 1 FOR THE FI-
NAL ITERATION PRIOR TO COMPARISON OF MASS FLOW RATES
IF(NI.GT.1.AND.NI.LT.NDIV) GO TO 440
IF(NI.EQ.NDIV) GO TO 230
IF(NDEL.EQ.1) GO TO 210
TAVERG=T(ICCR(NENTRF,2))
CONTINUE
IF(NDELFN.EC.1) GO TO 220
210
C
C
C
C
C
DELMAX-----FILM THICKNESS AT CONDENSER END OF PIPE(X=0.0)
DERIV-----DERIVATIVE OF FILM THICKNESS WITH RESPECT TO X AT
OVERFALL
CALL DELCRV(NDIV,NI,ITER,IFLUID,CL1,TSAT,TAVERG,RBASEI,OMEGA,DELMAX,
&X,DELSTR,DERIV,TAVG,NDEL,TALFA,CALFA,ZZERO,BFIN,EPS,T,CRITDL
&T,TBSFIN,NTERM)
IF(NTERM.EQ.1)GO TO 240
CONTINUE
GO TO 440
220
C
C
C
CONTINUE
GO TO 440
230
C
C
C
C
C
OVRDEL-----FILM THICKNESS AT THE OVERFALL
FLOMAS-----MASS FLOW RATE DETERMINED AT THE OVERFALL
OVRDEL=0.25(CO*DELMAX
FLOMAS=(((-2.000*PI*RHOF(NDIV)**2*OMEGA**2*R(NDIV)**2)/UF(NDIV)))*
&DERIV*(0.25(CO*DELMAX)**3/3.000
FLOW1=(RHCF(NDIV)**2*OMEGA**2*R(NDIV))/(3.000*UF(NDIV))
FLOW=FLOW1*(-1.000*OVRDEL**2*DERIV)*(EPS(NI)*OVRDEL+OVRDEL**2*
&TALFA)*ZFIN
IF(BFIN.NE.C.000)FLOMAS=FLOW
C
C
C
C
C
COMPARE MASS FLOW RATE AT OVERFALL TO CONDENSING MASS FLOW RATE,
IF NOT EQUAL, ADJUST DELMAX AND ITERATE AGAIN
UMFSAV=DELMF
DELSAV=DELMAX
DELMF=(DMTOT-FLOMAS)
DEFM=DABS(DELMF/DMTOT)
IF(NONCE.EQ.1)GO TO 240
IF(DEFM.GT.CRITDL) GO TO 250
CONTINUE
NSTOP=1
240

```



```

250 WRITTE(6,470)ITER
    WRITTE(6,480)ITER,FLOMAS,DELMF,DELSAV,DERIV,CVRDEL
    GO TO 440
    CONTINUE
    IF(ITER.GT.1) GO TO 260
    DEFMSV=CEFM
    DLMXSV=DELMAX
    CONTINUE
    IF(DEFM.GE.DEFMSV)GO TO 270
    DEFMSV=CEFM
    DLMXSV=DELMAX
    ITERSV=ITER
    CONTINUE
    IF(DELMF.GT.0.000) GO TO 330
    IF(BFIN.GT.0.000)GO TO 290
    IF(DEFM.LT.1.000)GO TO 280
    DELMAX=0.96500*DELMAX
    GO TO 320
    IF(DEFM.LT.0.1000) GO TO 290
    DELMAX=0.982500*DELMAX
    GO TO 320
    IF(DEFM.LT.0.0500) GO TO 300
    DELMAX=0.9912500*DELMAX
    GO TO 320
    IF(DEFM.LT.0.02500) GO TO 310
    DELMAX=0.99562500*DELMAX
    GO TO 320
    CONTINUE
    DELMAX=C.59785D0*DELMAX
    CONTINUE
    GO TO 350
    CONTINUE
    IF(BFIN.GT.0.000)GO TO 350
    IF(DEFM.LT.1.000)GO TO 340
    DELMAX=1.02000*DELMAX
    GO TO 380
    IF(DEFM.LT.0.1000) GO TO 350
    DELMAX=1.015000*DELMAX
    GO TO 380
    IF(DEFM.LT.0.0500)GO TO 360
    DELMAX=1.00187500*DELMAX
    GO TO 380
    IF(DEFM.LT.0.02500) GO TO 370
    DELMAX=1.00093700*DELMAX
    GO TO 380
    CONTINUE
    DELMAX=1.0004687500*DELMAX
    CONTINUE
360
370
380

```



```

390      CONTINUE
      IF((DELMF.GT.0.0D0.AND.DMFSAV.GT.0.0D0).OR.
&(DELMF.LT.0.0D0.AND.DMFSAV.LT.0.0D0)) GC TO 420
      IF(CELSAV.GT.DELMAX) GO TO 400
      DELMAX=DELSAV+0.5D0*(DELMAX-DELSAV)
      GO TO 410
400      CONTINUE
      DELMAX=DELMAX+0.5D0*(DELSAV-DELMAX)
410      CONTINUE
420      CONTINUE
      IF(ITRPRT.EC.0)GO TO 430
      WRITE(6,490)ITER,FLOMAS,DELMF,DELSAV,DERIV,OVRDEL
430      CONTINUE
      ITER=ITER+1
      NSTOP=0
      NDEL=0
      IF(ITER.EQ.ITERMX) NSTOP=1
      IF(ITER.EQ.ITERMX)WRITE(6,500)ITERMX,ITERSV,DEFMSV,DLMXSV
440      CONTINUE
      RETURN
450      FORMAT (1X,29HROOTS OF 4TH ORDER POLYNOMIAL,/,/6X,6HROOT 1,10X,6HRO
      LOT 2,0T 7X,6HROOT 3,10X,6HROOT 4,/)
460      FORMAT (4G15.7)
470      FORMAT (10X,'CONVERGENCE MET ON ITERATION NUMBER =',I5,/)
480      FORMAT (1X,'CN ITERATION NUMBER',I5,2X,'THE FOLLOWING CONDITIONS EX
&IST',/1X,'MASS FLOW RATE AT THE OVERFALL =',F15.5,2X,'LBM/HR',/1X,
&THE DIFFERENCE BETWEEN THE TWO CALCULATED MASS FLOW RATES IS',
&F15.8,/1X,'THE MAXIMUM FILM THICKNESS IS =',F15.8,/1X,'THE DERIVATI
&VE AT THE OVERFALL IS =',G15.8,/1X,'FILM THICKNESS AT OVERFALL =',
&F15.8,/)
490      FORMAT (1X,'CN ITERATION NUMBER',I5,2X,'THE FOLLOWING CONDITIONS EX
&IST',/1X,'MASS FLOW RATE AT THE OVERFALL =',F15.5,2X,'LBM/HR',/1X,
&THE DIFFERENCE BETWEEN THE TWO CALCULATED MASS FLOW RATES IS',
&F15.8,/1X,'THE MAXIMUM FILM THICKNESS IS =',F15.8,/1X,'THE DERIVATI
&VE AT THE OVERFALL IS =',G15.8,/1X,'FILM THICKNESS AT OVERFALL =',
&F15.8,/)
500      FORMAT (1X,'CONVERGENCE WAS NOT MET AFTER ',I5,2X,'ITERATIONS',/1X,
&HOWEVER, ON THE ',I5,2X,'ITERATION,THE MINIMUM REALTIVE DIFFERENC
&E WAS ACHIEVED',/1X,'THIS MINIMUM DIFFERENCE WAS EQUAL TO',D20.12,
&/1X,'THE DELMAX TO ACHIEVE THIS MINIMUM DIFFERENCE WAS',D20.12,/)
      END
      SUBROUTINE FORMAF FORMS STIFFNESS(A) AND FORCING(F)
      MATRICES
      SUBROUTINE FORMAF(A,F,NBAN)
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION A(200,50),F(200,1),B(3),C(3),EA(3,3)
      COMMON /GLOB1/ BOA,BFINI,CANGL,CLI,ETOEC,FANGL,HINF,QBTOT,RBASEI,R

```



```

&PM,R21,THICK1,TINF,TINTL,TSAT,ZFIN
COMMON /GLOB2/AOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
1ITDL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
2LOMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
3(100,50),QFNATOT,QINC(100),QINCSM(100),QSMTOT,QTFTOT,QTINC(100),QTQ
4IAL(100),QX(100),R(100),RHO(100),SALFA,SLNGTH(100),SPHI,SURFAR,TT(
5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TIRJF(100),THICK
6,UUF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NE,XILT,NEXTRT,NBSF
7IN,NCMTFI,NCMTRF,NDIV,NEL,NENTRF,NI,NP,DIFF,NPORIG,NPFCNV(10),NPFSY
8M(10),NPSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
DO 20 N=1,NSNP
F(N,1)=C.ODC
DO 10 MA=1,NBAN
A(N,MA)=0.010
CONTINUE
DO 120 IEL=1,NEL
IA=ICOR(1EL,1)
IB=ICOR(1EL,2)
IC=ICOR(1EL,3)
B(1)=Y(1B)-Y(1C)
B(2)=Y(1C)-Y(1A)
B(3)=Y(1A)-Y(1B)
C(1)=X(1C)-X(1B)
C(2)=X(1A)-X(1C)
C(3)=X(1B)-X(1A)

```

10
20

```

CALCULATE LENGTH DOMAIN (EL)
EL=DSQRT(C(3)**2+B(3)**2)

```

C
C
C
C
C

```

CALCULATE AREA DOMAIN
AS=CABS((B(1)*C(2)-B(2)*C(1))/2.0D0)
ESTABLISH HEAT TRANSFER COEFFICIENT
IF (1EL.LE.NEXTLT) GO TO 30

```

```

HC=0.0D0
GO TO 40
CONTINUE
HC=H(1EL)/Ck(NI)
CONTINUE

```

30
40
C
C
C

```

FORM STIFFNESS (A) MATRIX
DO 80 J=1,3
JJ=ICOR(1EL,J)
DO 70 K=1,3
KK=ICOR(1EL,K)
EA(J,K)=(B(J)*B(K)+C(J)*C(K))/(4*AS)

```



```

50 IF (HC*EQ.0.0D0) GO TO 60
   HEL=HC*EL/6.0D0
   IF (J.EQ.3) GO TO 60
   IF (K.EC.3) GO TO 60
   IF (J.EC.K) GO TO 50
   EA(J,K)=EA(J,K)+HEL
   GO TO 60
50 CONTINUE
   EA(J,K)=EA(J,K)+2*HEL
   IF (KK.LT.JJ) GC TC 70
   NW=KK-JJ+1
   A(JJ,NW)=A(JJ,NW)+EA(J,K)
   CONTINUE
   CONTINUE
C
C
C
90 FORM FORCING MATRIX(F)
   IF (IEL.GT.NENTRF) GO TO 90
   FE=HC*TSAT*EL/2.0D0
   GO TO 110
   IF (IEL.GT.NEXTLT) GO TO 100
   FE=HC*TINGF*EL/2.0D0
   GO TO 110
   CONTINUE
   FE=0.0D0
   CONTINUE
   F(IA,1)=F(IA,1)+FE
   F(IE,1)=F(IE,1)+FE
   CONTINUE
   RETURN
   END
C
C
C
SUBROUTINE BANDEC SOLVES FOR T MATRIX
SUBROUTINE EANDEC (A,F,NEQ,MAXB,NVEC)
IMPLICIT REAL*8(A-H,O-Z)
DIMENSION A(200,50),F(200,1)
COMMON /GLOBE/ BOA,BFINI,CANGL,CL1,ETOEO,FANGL,HINF,QBTOT,RBASEI,R
&PM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
COMMON /GLOBE2/ AFOVAS,AMTUT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
1ITDL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
2LOMAS,H(200),HELMNT(100,50),HFG,UMEGA,PHI,PI,QBI,QBINC(100),QELMNT
3(100,50),QFNTOT,QINC(100),QINCSM(100),QSMFOT,QTFBOT,QTINC(100),CTU
4TAL(100),QX(100),R(100),RHCF(100),SALFA,SLNGH(100),SPHI,SURFAR,TC
5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TIROF(100),THICK
6UF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NEXTLT,NEXTRT,NBSF
7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NI,NPDIFF,NPORIG,NPFCNV(10),NPFSY
8M(10),NPSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
   LOOP=NEC-1

```



```

DO 20 I=1,LCP
MB=I+1
NB=MINO(I+MAXB-1,NEQ)
DO 20 J=MB,NB
L=J+2-ME
U=A(I,L)/A(I,1)
DO 10 M=1,NVEC
F(J,MM)=F(J,MM)-D*F(I,MM)
MM=MINO(MAXE-L+1,NEQ-J+1)
DO 20 K=1,MM
NN=L+K-1
A(J,K)=A(J,K)-D*A(I,NN)
DO 30 I=1,NVEC
F(NEQ,I)=F(NEQ,I)/A(NEQ,1)
DO 50 I=2,NEQ
J=NEQ-I+1
K=MINO(NEQ-J+1,MAXB)
DO 50 M=1,NVEC
DO 40 L=2,K
MB=J+L-1
F(J,MM)=F(J,MM)-A(J,L)*F(MB,MM)
F(J,MM)=F(J,MM)/A(J,1)
RETURN
END

SUBROUTINE HTCOEF DETERMINES HEAT TRANSFER COEFFICIENTS

SUBROUTINE HTCOEF(AA1,BB1,CPI)
IMPLICIT REAL*8(A-H,O-Z)
DIMENSION HZ(5)
COMMON /GLOB1/ BOA,BFINI,CANGL,CLI,ETOEG,FANGL,HINF,QBTOT,RBASEI,R
&PM,R21,THICKI,TINF,TINTL,TSAT,ZFIN
COMMON /GLOB2/AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
1ITDL,CW(100),DELSTR(100),DELX,DMTGT,ELMNT(100,50),EPS(100),EZERO,F
2LOMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
3(100,50),QFNTOT,QINC(100),QINCSM(100),QSMFOT,QTFOT,QTINC(100),QTO
4TAL(100),QX(100),RHOF(100),SALFA,SLNGTH(100),SPHI,SURFAR,T(
5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TIROF(100),THICK
6,UF(100),X(100),Y(100),Z(100),ZZERG,ICCR(200,3),NEXTLT,NEXTRI,NBSF
7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NI,NPDIFF,NPORIG,NPFCNV(10),NPFSY
8M(10),NPSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
COMMON/RECT/NFN TIP

AVERAGE ELEMENT CONVECTIVE HEAT TRANSFER COEFFICIENT ALONG FIN
SURFACE
ZSTAR-----DISTANCE ALONG FIN SURFACE FROM APEX TO SURFACE OF
TROUGH CONDENSATE
DELZ-----AVERAGE CONDENSATE FILM THICKNESS ON CONVECTIVE
BOUNDARY

```



```

AZ-----Z COORDINATE OF UPPER NODAL POINT OF ELEMENT
BZ-----Z COORDINATE OF LOWER NODAL POINT OF ELEMENT
AK-----SUBDIVISION OF ELEMENT LENGTH(5 FOR FIRST ELEMENT AND
          4 FOR OTHER ELEMENTS)
HAZ-----WEIGHTED THERMAL CONDUCTIVITY CF TROUGH CONDENSATE
          (BTU/HR-FT-DEGF)
HAC-----WEIGHTED THERMAL CONDUCTIVITY CF FILM CONDENSATE FOR
          ELEMENT IN CONTACT WITH BOTH TROUGH AND FILM CONDEN-
          SATE(BTU/HR-FT-DEGF)
CONH-----AVERAGE HEAT TRANSFER COEFFICIENT(BTU/HR-FT2-DEGF)
HZ-----LOCAL HEAT TRANSFER COEFFICIENT(BTU/HR-FT2-DEGF)
DELA-----FILM THICKNESS AT UPPER NODAL POINT OF ELEMENT(FEET)
DELB-----FILM THICKNESS AT LOWER NODAL POINT OF ELEMENT(FEET)
H-----AVERAGE HEAT TRANSFER COEFFICIENT OF THE SURFACE OF
          CONVECTIVE BOUNDARY ELEMENTS(BTU/HR-FT2-DEGF)

```

```

IF (BFIN.EQ.0.000) GO TO 140

```

```

DETERMINE HEAT TRANSFER COEFFICIENT FOR HORIZONTAL SURFACE OF
FIN.

```

```

NOTE:FOR EASE OF ANALYSIS, ASSUME SURFACE IS INSULATED,I.E.

```

```

DO 5 IEL=1,NCMREC
  H(IEI)=0C.000
CONTINUE

```

```

DETERMINE HEAT TRANSFER COEFFICIENTS FOR ELEMENTS ON FIN
VERTICAL SURFACE
ZSTAR=ZZERO-(DELSTR(NI)/CALFA)
HDEN=(-1.0DC*(AA1*ZSTAR**3/3.000+BB1*ZSTAR**2/2.000))+ZSTAR*(TSAT-
T(1))
CONST=RHOF(NI)**2*OMEGA**2*HFG*CPHI*CALFA*(NI)
DELA=DAES(4*CF(NI)*UF(NI)*HDEN/CONST)**0.2500
HAC=0.000
INP=1

```

```

DO 130 IEL=NFNTIP,NBSFIN
  AZ=Z(INP)
  BZ=Z(INP+1)
  IF (ZSTAR.LE.BZ) GO TO 10
  GO TO 30
  IF (HAC.NE.C.000) GO TO 110
  IF (IEL.NE.NFNTIP) GO TO 20
  AK=(ZSTAR-AZ)/5.000
  ZZ=AK
  GO TO 50

```

10


```

20  CONTINUE
    AK=(ZSTAR-AZ)/4.0D0
    ZZ=AZ
    GO TO 50
30  IF (IEL.NE.NFNTIP) GO TO 40
    AK=(BZ-AZ)/5.0DC
    ZZ=AK
    GO TO 50
40  CONTINUE
    AK=(BZ-AZ)/4.0D0
    ZZ=AZ
50  CONTINUE
    ZEL=4.0C0*AK
    DO 60 NH=1, ZEL
    HDEN=(-1.0DC*(AA1*ZZ**3/3.0D0+BB1*ZZ**2/2.0D0))+ZZ*(TSAI-T(1))
    HZ(NH)=DABS(CF(NI)**3*CONST/(4.0D0*UF(NI)*HCEN))**0.25D0
    ZZ=ZZ+AK
    CONTINUE
60  CONH=AK*(HZ(1)+4.0D0*HZ(2)+2.0D0*HZ(3)+4.0D0*HZ(4)+HZ(5))/(3.0D0*Z
    IEL)
    IF (ZSTAR.LE.BZ) GO TO 70
    H(IEL)=CCNH
    GO TO 120
70  CONTINUE
    HAZ=CCNH*(ZSTAR-AZ)
80  CONTINUE
    AZZ=DELSTR(NI)
    DELB=(BZ-ZSTAR)*AZZ/(ZZERO-ZSTAR)
    IF (ZSTAR.EC.BZ) DELB=DELA
    DELZ=(DELA+DELB)/2.0D0
    IF (AZ.LT.ZSTAR) GO TO 90
    HAC=(BZ-AZ)*CF(NI)/DELZ
    GO TO 100
90  CONTINUE
    HAC=(BZ-ZSTAR)*CF(NI)/DELZ
100 CONTINUE
    H(IEL)=(HAZ+HAC)/(BZ-AZ)
    GO TO 120
110 CONTINUE
    DELA=DELB
    HAZ=0.0C0
    GO TO 80
111 CONTINUE
120 INP=INP+1
130 CONTINUE
140 C

```



```
C C
DETERMINE HEAT TRANSFER COEFFICIENT FOR ELEMENTS IN TROUGH
NEXT=NBSFIN+1
IF (BFIN.EQ.0.0DO) NEXT=1
DO 150 IEL=NEXT,NENTRF
H(IEI)=CF(NII)/DELSTR(NI)
CONTINUE
150 C C
DEFINE HEAT TRANSFER COEFFICIENT ALONG BOTTOM(OUTSIDE) SURFACE
DO 160 ICL=NEXTIRT,NEXILT
H(ICL)=FINF
CONTINUE
RETURN
END
160 C C
SUBROUTINE HTCALC PERFORMS HEAT TRANSFER CALCULATIONS
SUBROUTINE FTICALC
IMPLICIT REAL*8(A-H,D-Z)
COMMON /GLOBAL/ EOA,BFINI,CANGL,CLI,ETOEC,FANGL,HINF,QBTOT,RBASEI,K
&PM,R2I,THICKI,TINF,TINTL,TISAT,ZFIN
COMMON /GLOBE2/AFOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
1 ITDL,CW(100),DELMSTR(100),DELX,DMTOT,PHI,PI,QBI,QBINC(100),QELMNT
2 LOMAS,H(200),HELMNT(100,50),HF,G,OMEGA,PSI,QFTOT,QTINC(100),QTU
3 (100,50),QFNTOI,QINCI(100),QINCSM(100),QSMTOI,QTFOTI,SPHI,SURFAR,I(
4 TAL(100),QX(100),R(100),RHOF(100),SALFA,SANGTH(100),TBRC(100),TIRUF(100),THICK
5 I(100),TALFA,TBL(100),TBM(100),TZERO,IICR(200,3),NEXILT,NEXTIRT,NBSF
6 UF(100),X(100),Y(100),Z(100),ZZERO,NENTRF,NI,NPDIFF,NPORIG,NPFCNV(I,O),NPFSY
7 IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NRWFIN,NRWREC,NSNP,NCMREC,NSOLVE
8 M(10),NPFSMBS,NRFIN,NRWFIN,NRWREC,NSNP,NCMREC,NSOLVE
HEAT RATE CALCULATIONS
NOTE FOR ALL HEAT CALCULATIONS THE FOLLOWING VARIABLES ARE
DEFINED:
KA-----FIRST NODAL POINT OF ELEMENT
KB-----SECOND NODAL POINT OF ELEMENT
XQEL-----DISTANCE BETWEEN NODAL POINTS IN X DIRECTION
YQEL-----DISTANCE BETWEEN NODAL POINTS IN Y DIRECTION
ELM-----ELEMENT LENGTH ON CONVECTIVE BOUNDARY
ELMNT-----ELEMENT LENGTH ON CONVECTIVE BOUNDARY
HELMNT-----HEAT TRANSFER COEFFICIENT FOR CONVECTIVE BOUNDARY
QE LMNT-----ELEMENT NT
-----ELEMENT RATE
-----HEAT RATE
-----HEAT RATE PER UNIT LENGTH FOR CONVECTIVE BOUNDARY
HEAT RATE INTO FIN SECTION
-----HEAT RATE INTO ELEMENTS ON CONVECTIVE BOUNDARY PER
```



```

C C C C C C C C
      INCREMENTAL LENGTH(BTU/HR-FT)
      INTO ONE FIN SECTION FOR A GIVEN IN-
      CREMENT(BTU/HR)
      INTO ALL FIN SECTIONS FOR A GIVEN IN-
      CREMENT(BTU/HR)
      INTO ALL FIN SECTIONS FOR NI INCRE-
      MENTS(BTU/HR)
      QINC-----TOTAL HEAT RATE
      QET-----TOTAL HEAT RATE
      QFNTOT-----TOTAL HEAT RATE
      IF (BFIN.EQ.0.0D0) GO TO 30
      QEL=0.0D0
      DO 10 IQEL=1,NBSFIN
      KA=ICOR(IQEL,1)
      KB=ICOR(IQEL,2)
      XQEL=X(KA)-X(KB)
      YQEL=Y(KA)-Y(KB)
      ELM=DSQRT(XQEL**2+YQEL**2)
      ELMNT(NI,IQEL)=ELM
      HELMNT(NI,IQEL)=H(IQEL)
      QELMNT(NI,IQEL)=(2*TSAT-T(KA)-T(KB))*ELM*(H(IQEL)/2.0D0
      QEL=QEL+(2*TSAT-T(KA)-T(KB))*ELM*(H(IQEL)/2.0D0
      CONTINUE
      QINC(NI)=QEL*DELX
      QET=QEL*DELX*ZFIN*2
      QFNTOT=QFNTCT+QET

```

10

```

      HEAT RATE INTO TROUGH
      QTRF-----HEAT RATE INTO TROUGH ELEMENTS IN CONVECTIVE BOUNDARY
      PER INCREMENTAL WIDTH (BTU/HR-FT)
      QTRF-----TOTAL HEAT RATE INTO THE TROUGH SECTION FOR A GIVEN
      INCREMENT(BTU/HR)
      QTRF-----TOTAL HEAT RATE INTO ALL TROUGH SECTIONS OF A GIVEN
      INCREMENT(BTU/HR)
      QTRFOT-----TOTAL HEAT RATE INTO ALL TROUGH SECTIONS FOR IN IN-
      CREMENTS(BTU/HR)
      QTRF=0.0D0
      NEXT=NBSFIN+1
      DO 20 IQEL=NEXT,NENTRF
      KA=ICOR(IQEL,1)
      KB=ICOR(IQEL,2)
      XQEL=X(KA)-X(KB)
      YQEL=Y(KA)-Y(KB)
      ELM=DSQRT(XQEL**2+YQEL**2)
      ELMNT(NI,IQEL)=ELM
      HELMNT(NI,IQEL)=H(IQEL)
      QELMNT(NI,IQEL)=(2*TSAT-T(KA)-T(KB))*ELM*(H(IQEL)/2.0D0
      QTRF=QTRF+(2*TSAT-T(KA)-T(KB))*ELM*(H(IQEL)/2.0D0

```



```

20  CONTINUE
    QTINC(NI)=QTRF*DELX
    QTRFT=QTRF*CELEX*ZFIN*2.000
    QTFTOT=CTFTCT+QTRFT
C
30  QINCSM(NI)=C.000
    GO TO 50
CONTINUE
    QELSM=0.000
    DO 40 ICEL=1,NENTRF
    KA=ICOR(ICEL,1)
    KB=ICOR(ICEL,2)
    XQEL=X(KA)-X(KB)
    YQEL=Y(KA)-Y(KB)
    ELM=DSCRT(XQEL**2+YQEL**2)
    ELMNT(NI,ICEL)=ELM
    HELMNT(NI,ICEL)=H(ICEL)
    QELSM=QELSM+(2*TSAT-T(KA)-T(KB))*ELM*H(ICEL)/2.000
    QINCSM(NI)=QINCSM(NI)+QELSM
CONTINUE
40  QINCSM(NI)=QELSM*DELX
    QSMT=QELSM*CELEX*360.000
    QSMTOT=QSMTCT+QSMT
CONTINUE
    TOTAL HEAT RATE INTO A SECTION THROUGH FIN AND TROUGH
    QTOTAL=-----TOTAL HEAT RATE INTO A GIVEN SECTION THROUGH FIN AND
    TROUGH IN A GIVEN INCREMENT(BTU/HR)
    QTOTAL(NI)=(QINC(NI)+QTINC(NI)+QINCSM(NI))
C
    HEAT RATE FROM BOTTOM TO AMBIENT
    QBI=-----HEAT RATE FROM BOTTOM ELEMENTS OF SECTION PER INCRE-
    MENTAL LENGTH(BTU/HR-FT)
    QBINC=-----TOTAL HEAT RATE OUT OF BOTTOM OF SECTION FOR A
    INCREMENT(BTU/HR)
    QBI=-----TOTAL HEAT RATE OUT OF BOTTOM OF ALL SECTIONS IN
    A GIVEN INCREMENT(BTU/HR)
    QBTTOT=-----TOTAL HEAT RATE OUT OF HEAT PIPE FOR NI INCREMENTS
    (BTU/HR)
C
    QBI=0.000
    DO 60 IQEL=NEXTRT,NEXTLT
    KA=ICOR(ICEL,1)
    KB=ICOR(ICEL,2)
    XQEL=X(KA)-X(KB)
    YQEL=Y(KA)-Y(KB)

```



```

60      ELM=DSQRT(XCEL**2+YQEL**2)
      ELMNT(NI,IQEL)=ELM
      HELMNT(NI,IQEL)=H(IQEL)
      QELMNT(NI,IQEL)=(T(KA)+T(KB)-2*TINF)*ELM*(IQEL)/2.0D0
      QBI=QBI+(T(KA)+T(KB)-2*TINF)*ELM*(IQEL)/2.0D0
      CONTINUE
      QBINC(NI)=QBI*DELX
      QB=QBI*CELEX*ZFIN*2.0D0
      IF(BFIN.NE.C.0D0) GO TO 70
      QB=QBI*CELEX*360.0D0
      QX(NI)=QBINC(NI)*360.0D0
      CONTINUE
      QBTOT=QBTOT+QB
      RETURN
      END
C      SUBROUTINE SIUNIT CONVERTS A DIMENSIONAL QUANTITIES TO SI UNITS
C      AND CUTPLTS THESE QUANTITIES
C
C      SUBROUTINE SIUNIT(TI,TPLT,XPLT,YPLT,NFLAG1,NFLAG2,NFLAG3)
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION TPLT(1,1),XPLT(1,1),YPLT(1,1),TI(1)
      COMMON /GLOB1/ BOA,BFIN,CANGL,CLI,EJOEO,FANGL,HINF,QBTOT,RBASEI,R
      GPM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
      COMMON /GLOB2/ AFOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
      ITDL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
      2LOMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
      3(100,50),QFNTOI,QINC(100),QINCSM(100),QSMTOI,QTFOT,QTINC(100),QTD
      4TAL(100),QX(100),R(100),RHOF(100),SALFA,SLNGTH(100),SPHI,SURFAR,TH
      5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TTRUF(100),THICK
      6,UUF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NEXTLT,NEXTRI,NBSF
      7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NIPDIFF,NPDIRIG,NPFCNV(10),NPFSY
      8M(10),NPSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
      SUBROUTINE SIUNIT(TI,TPLT,XPLT,YPLT,NFLAG1,NFLAG2,NFLAG3)
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION TPLT(1,1),XPLT(1,1),YPLT(1,1),TI(1)
      COMMON /GLOB1/ BOA,BFIN,CANGL,CLI,EJOEO,FANGL,HINF,QBTOT,RBASEI,R
      GPM,R2I,THICKI,TINF,TINTL,TSAT,ZFIN
      COMMON /GLOB2/ AFOVAS,AMTOT(100),BFIN,CALFA,CF(100),CP(100),CRIT,CR
      ITDL,CW(100),DELSTR(100),DELX,DMTOT,ELMNT(100,50),EPS(100),EZERO,F
      2LOMAS,H(200),HELMNT(100,50),HFG,OMEGA,PHI,PI,QBI,QBINC(100),QELMNT
      3(100,50),QFNTOI,QINC(100),QINCSM(100),QSMTOI,QTFOT,QTINC(100),QTD
      4TAL(100),QX(100),R(100),RHOF(100),SALFA,SLNGTH(100),SPHI,SURFAR,TH
      5100),TALFA,TBL(100),TBM(100),TBR(100),TBSFIN(100),TTRUF(100),THICK
      6,UUF(100),X(100),Y(100),Z(100),ZZERO,ICOR(200,3),NEXTLT,NEXTRI,NBSF
      7IN,NCMTRI,NCMTRF,NDIV,NEL,NENTRF,NIPDIFF,NPDIRIG,NPFCNV(10),NPFSY
      8M(10),NPSMBS,NRFIN,NRWFIN,NRWTRF,NSNP,NCMREC,NRWREC,NCOL,NSOLVE
      PRINT HEADR
      WRITE (6,17C)
      CONVERT DIMENSIONAL INPUT PARAMETERS TO SI UNITS
      CLI=CLI/39.37007D0
      RBASEI=RBASEI/39.37007D0
      THICKI=THICKI/39.37007D0
      BFINI=BFINI/39.37007D0

```



```

HINF=HINF/.176225D0
TINTL=(TINTL-32.0D0)*(5.0D0/9.0D0)
TSAT=(TSAT-32.0D0)*(5.0D0/9.0D0)
TINF=(TINF-32.0D0)*(5.0D0/9.0D0)

      C C C C

      OUTPUT INPUT DATA IN SI UNITS

      WRITE (6,18C) CLI,RBASEI,THICKI,BFINI,CANGL
      WRITE (6,19C) FANGL,ETCEO
      WRITE (6,20C) CRIT,CRITDL
      WRITE (6,21C) RPM,TINTL,TSAT,TINF,HINF

      C C C C

      PRINT HEADER FOR CALCULATED RESULTS
      WRITE (6,22C)

      C C C C

      OUTPUT FIN GEOMETRY PARAMETERS
      SURFAR=SURFAR*.3048D0
      WRITE (6,33C) ZFIN,FANGL,SURFAR,ETOEO,BGA,AFOVAS

      C C C C

      OUTPUT LATENT HEAT OF VAPORIZATION FOR A GIVEN TSAT

      HFG=HFG*2.324444D0
      WRITE (6,34C) HFG,TSAT

      C C C C C

      OUTPUT HEAT RATE INTO TROUGH,HEAT RATE INTO FIN,TOTAL HEAT
      RATE IN AND HEAT RATE OUT FOR A SINGLE SECTION FOR EACH
      INCFEMENT

      DO 10 NR=1,NDIV
      QINC(NR)=QINC(NR)/3.412322D0
      QTINC(NR)=QTINC(NR)/3.412322D0
      QTOTAL(NR)=QTOTAL(NR)/3.412322D0
      QBINC(NR)=QBINC(NR)/3.412322D0
      QINCSM(NR)=QINCSM(NR)/3.412322D0
      QX(NR)=QX(NR)/3.412322D0
      CONTINUE
      DMTOT=DMTOT/7956.639D0
      FLOMAS=FLOMAS/7956.639D0
      IF (BFIN.EQ.0.0D0) GO TO 30

      C C C C C

      OUTPUT HEAT RATE INTO TROUGH,HEAT RATE INTO FIN,TOTAL HEAT
      RATE IN AND HEAT RATE OUT FOR A SINGLE SECTION FOR EACH
      INCFEMENT

```



```

20 WRITE (6,23C)
DO 20 NR=1,NDIV
WRITE (6,24C) NR,QINC(NR),QTINC(NR),QTOTAL(NR),QBINC(NR)
CONTINUE
30 GO TO 8C
CONTINUE
IF (NSCLVE.NE.1) GO TO 4C
40 GO TO 7C
IF (NSCLVE.NE.2) GO TO 5C
WRITE (6,26C)
50 GO TO 7C
IF (NSCLVE.NE.3) GO TO 6C
WRITE (6,27C)
60 GO TO 7C
CONTINUE
WRITE (6,28C)
70 CONTINUE
WRITE (6,29C)
DO 80 NR=1,NDIV
WRITE (6,30C) NR,QINCSM(NR),QBINC(NR),QX(NR)
80 CONTINUE

      OUTPUT HEAT RATE INTO FIN, HEAT RATE INTO TROUGH, AND HEAT
      RATE OUT BOTTOM AND TOTAL MASS FLOW RATE FOR A GIVEN SET
      OF INPUT CONDITIONS

      OUTPUT HEAT RATE INTO FIN, HEAT RATE INTO TROUGH, AND HEAT
      RATE OUT BOTTOM AND TOTAL MASS FLOW RATE FOR A GIVEN SET
      OF INPUT CONDITIONS

QFNTOT=QFNTCT/3.412322D0
QTFTOT=QTFTCT/3.412322D0
QBTOT=QBTCT/3.412322D0
QSMTOT=QSMCT/3.412322D0
IF (BFIN.EQ.0.0D0) GO TO 90
WRITE (6,31C) QFNTOT,QTFTOT,QBTOT,DMTOT
90 GO TO 1C0
IF (PHI.EQ.0.0D0) GO TO 95
WRITE (6,315) QSMTOT,QBTCT,DMTOT
95 GO TO 1C0
CONTINUE
WRITE (6,32C) QSMTOT,QBTOT,DMTOT,FLCMAS
100 CONTINUE

      OUTPUT INCREMENTALLY VARYING PROPERTIES

WRITE (6,35C)
DO 110 NR=1,NDIV

```



```

110 CF(NR)=CF(NR)*1.729577D0
C CW(NR)=CW(NR)*1.729577D0
C UF(NR)=UF(NR)/2419.088D0
RHOF(NR)=RHCF(NR)/.0624279D0
WRITE (6,36C) NR,CF(NR),CW(NR),UF(NR),RHOF(NR)
CONTINUE

      OUTPUT INCREMENTALLY VARYING PARAMETERS

IF(BFIN.EQ.C.0D0) GC TO 125
WRITE (6,37C)
DO 120 NR=1,NDIV
DELSTR(NR)=DELSTR(NR)*.3048D0
EPS(NR)=EPS(NR)*.3048D0
R(NR)=R(NR)*.3048D0
AMTCT(NR)=AMTOT(NR)/7936.639D0
WRITE (6,38C) NR,DELSTR(NR),EPS(NR),R(NR),AMTOT(NR)
CONTINUE
GO TO 128
CONTINUE
WRITE (6,375)
DO 128 NR=1,NDIV
DELSTR(NR)=DELSTR(NR)*.3048D0
SLNGTH(NR)=SLNGTH(NR)*.3048D0
R(NR)=R(NR)*.3048D0
AMTCT(NR)=AMTOT(NR)/7936.639D0
WRITE (6,380) NR,DELSTR(NR),SLNGTH(NR),R(NR),AMTOT(NR)
CONTINUE

128      OUTPUT MAJOR TEMPERATURES FOR EACH INCREMENT
C
C
IF(BFIN.EQ.C.0D0) GC TO 135
WRITE (6,39C)
GO TO 138
CONTINUE
WRITE (6,395)
CONTINUE
DO 130 NR=1,NDIV
T1(NR)=(T1(NR)-32.0D0)*(5.0D0/9.0D0)
TBR(NR)=(TBR(NR)-32.0D0)*(5.0D0/9.0D0)
TBM(NR)=(TBM(NR)-32.0D0)*(5.0D0/9.0D0)
TBL(NR)=(TBL(NR)-32.0D0)*(5.0D0/9.0D0)
TBSFIN(NR)=(TBSFIN(NR)-32.0D0)*(5.0D0/9.0D0)
TTROF(NR)=(TTROF(NR)-32.0D0)*(5.0D0/9.0D0)
WRITE (6,400) NR,TBR(NR),TBM(NR),T1(NR),TBSFIN(NR),TTROF(NR)
CONTINUE

130      OUTPUT NCAL PUINT X AND Y COORDINATES AND FINAL TEMPER-
C
C

```


TURE FOR EACH NODAL POINT FOR INCREMENT OF INTEREST

```

WRITE (6,41C)
DO 140 I=NFLAG1,NFLAG2,NFLAG3
WRITE (6,42C) I
WRITE (6,43C)
DO 140 NP=1,NSNP
XPLT(I,NP)=XPLT(I,NP)*.3048D0
YPLT(I,NP)=YPLT(I,NP)*.3048D0
TPLT(I,NP)=(TPLT(I,NP)-32.0D0)*(5.0D0/9.0D0)
WRITE (6,44C) NP,XPLT(I,NP),YPLT(I,NP),TPLT(I,NP)
CONTINUE

```

140
C
C
C
C
C

OUTPUT ELEMENT LENGTH, HEAT TRANSFER COEFFICIENT AND HEAT
RATE PER UNIT LENGTH FOR CONVECTIVE BOUNDARY ELEMENTS FOR
INCREMENTS OF INTEREST

```

WRITE (6,45C)
DO 150 IQEL=1,NEXTLT
ELMNT(I,IQEL)=ELMNT(I,IQEL)*.3048D0
HELMNT(I,IQEL)=HELMNT(I,IQEL)*5.674561DC
QELMNT(I,IQEL)=QELMNT(I,IQEL)/1.040076D0
WRITE (6,46C) IQEL,ELMNT(I,IQEL),HELMNT(I,IQEL),QELMNT(I,IQEL)
CONTINUE
CONTINUE

```

150
160
C
C
C

OUTPUT FORMAT

```

RETURN
FORMAT (1H1,///30X,16HINPUT PARAMETERS,14X,48HALL DIMENSIONAL QU
PARAMETERS ARE GIVEN IN SI UNITS,///)
FORMAT (1X,18HCONDENSER LENGTH =,6X,G10.5,2X,6HMETERS,1X,16HMINIM
LUM RADIUS =,8X,G10.5,2X,7HMETERS,1X,16HWALL THICKNESS =,8X,G10.5
2,1X,7HMETERS,1X,12HFIN HEIGHT =,12X,G10.5,1X,7HMETERS,1X,22HCO
NDENSER HALF ANGLE =,1X,G10.5,2X,8H DEGREES//)
FORMAT (10X,14HFIN PARAMETERS,1X,16HFIN HALF ANGLE =,610.5,8H DEG
REES,1X,38H RATIO OF TROUGH WIDTH TO BASE OF FIN =,610.5,///)
FORMAT (1X,16HTEMPERATURE COVERAGE CRITERION =,10X,G10.5,1X,16HMASS FL
OW CONVERGENCE CRITERION =,610.5,1X,16HNOT PIPE ANALYSIS,///)
FORMAT (10X,20HOPERATING PARAMETERS,1X,5H RPM =,10X,G10.5,23H REVOLUT
IONS PER MINUTE,1X,30H SATURATION TEMPERATURE ESTIMATE =,2X,G10.5,
210H DEGREES C,1X,24H SATURATION TEMPERATURE =,8X,G10.5,10H DEGREES
3C,1X,22H EXTERNAL TEMPERATURE =,10X,G10.5,1X,9H DEGREES C,1X,36H
4 EXTERNAL HEAT TRANSFER COEFFICIENT =,610.5,10HW/M2-DEG K,///)
FORMAT (1H1,25X,18H CALCULATED RESULTS,125X,19H RESULTS IN SI UNITS,
1///)

```

220
230

```

FORMAT (1H0,1X,36H HEAT RATE SUMMARY FOR A UNIT SECTION,1X,11HELEM

```



```

1ENT NR.,3X,10HHEAT RATE,5X,10HHEAT RATE,6X,9HHEAT RATE,5X,9HHEAT
2RATE,7X,8HINTO FIN,6X,11HINTO TROUGH,6X,8HTOTAL IN,6X,10HOUT BO
3TTOM,7X,8H(WATTS),7X,8H(WATTS),8X,8H(WATTS),6X,8H(WATTS),7X
FORMAT(1X,15,10X,G10.5,3(5X,G10.5))//
FORMAT(1X,RESULTS BASED ON BALLBACK EQUATION FOR TROUGH THICKNESS
&,///)
FORMAT(1X,RESULTS BASED ON DANIELS EQUATION FOR TROUGH THICKNESS
&,1X,SIMPLIFIED FORM-FUNCTION OF SHERWOOD NUMBER ONLY,///)
FORMAT(1X,RESULTS BASED ON DANIELS EQUATION FOR TROUGH THICKNESS
&,1X,FULL FORM,INCLUDES DRAG TERMS,///)
FORMAT(1H0,15X,36HHEAT RATE SUMMARY FOR A UNIT SECTION,7X,10HIN
1CREMENT,3X,10HHEAT RATE,5X,10HHEAT RATE,8X,TOTAL HEAT RATE,7X
22X,12HINTO SECTION,4X,14HOUT OF SECTION,9X,FOR ALL SECTIONS,7X
38H(WATTS),7X,8H(WATTS),8X,IN THE INCREMENT,///)
FORMAT(1X,15,10X,G10.5,5X,G10.5,9X,G10.5)
FORMAT(1H0,27HTOTAL HEAT RATE INTO FINS,9X,F10.3,7H WATTS,7X
1,27HTOTAL HEAT RATE INTO TROUGH,8X,F10.3,7H WATTS,7X,32HTOTAL HE
2AT RATE,3X,G10.5,7H KG/S,///)
FORMAT(1H0,1X,31HTOTAL HEAT RATE INTO HEAT PIPE,3X,F10.3,7H WATT
3OW RATE,3X,G10.5,7H KG/S,///)
FORMAT(1H0,1X,32HTOTAL HEAT RATE OUT OF HEAT PIPE,3X,F10.3,7H WATT
1S,///1X,20HTOTAL MASS FLOW RATE,3X,G10.5,7H KG/M3,///)
FORMAT(1H0,1X,31HTOTAL HEAT RATE INTO HEAT PIPE,3X,F10.3,7H WATT
1S,///1X,32HTOTAL HEAT RATE OUT OF HEAT PIPE,3X,F10.3,7H WATTS,///
2/1X,32HTOTAL HEAT RATE BASED ON HEAT RATE DIVIDED BY HFG,7X,F10.3
3,3X,KG/SEC,7X,1X,TOTAL MASS FLOW RATE AT OVERFALL,7X,F10.3,3X,KG/S
4EC,///)
FORMAT(10X,23HFIN GEOMETRY PARAMETERS,7X,9HNR FINS =,5X,G10.5,7X
1X,14HFIN HALF ANGLE,5X,G10.5,5X,8HDEGREES,7X,32HFIN SURFACE AREA
2PER UNIT LENGTH,01X,G10.5,01X,12HMETER2/METER,7X,41HRAATIO OF TRO
3UGH WIDTH TO FIN BASE WIDTH =,01X,G10.5,7X,32HRAATIO OF FIN HEIGH
4T TO FIN BASE =,01X,G10.5,7X,42HRAATIO OF FIN SURFACE AREA TO SMOU
5TH AREA =,01X,G10.5,7X)
FORMAT(1H0,1X,29HLATENT HEAT OF VAPORIZATION =,G10.5,8H KJ/KG,7X
1X,20HFOR SATURATION TEMP =,G10.5,9HDEGREES C,///)
FORMAT(1H1,10X,43HFLUID AND MATERIAL PROPERTIES PER INCREMENT,7X
1,9HINCREMENT,7X,5HFLUID,15X,4HWALL,15X,SHVIScosity,12X,7HDENSITY,1
25X,12HCCNCUCTIVITY,7X,12HCCNCUCTIVITY,7X,5HW/M-DEG K,10X,9HW/M-D
3EG K,13X,6HN-S/M2,14X,5HKG/M3,7X)
FORMAT(1X,15,4(10X,G10.5))
FORMAT(1H1,10X,32HVARIOUS PARAMETERS PER INCREMENT,7X,10HINCREME
1NT,6X,8HDEL STAR,10X,12HTROUGH WIDTH,8X,14HMINIMUM RADIUS,6X,14HM
2ASS FLOW RATE,7X,16X,6HMETERS,14X,6HMETERS,14X,6H KG/S
3,///)
FORMAT(1H1,10X,32HVARIOUS PARAMETERS PER INCREMENT,7X,10HINCREME
1NT,6X,8HDEL STAR,10X,13HSECTION WIDTH,7X,14HMINIMUM RADIUS,6X,14H
2MASS FLOW RATE,7X,19X,6HMETERS,14X,6HMETERS,14X,6H KG/S
3,///)

```



```

380 FORMAT (1X, I5, 4(10X, G10.5)),
390 FORMAT (//, 17X, MAJOR TEMPERATURES PER UNIT SECTION FOR ALL INCREME
&NTS, //, 7X, EXTERNAL, 07X, EXTERNAL, 07X, EXTERNAL, 07X, INTERNAL,
&6X, INTERNAL, 07X, INTERNAL, 07X, LEFT, 08X, BELOW BASE, 08X, RIGHT
&, 09X, FIN TIP, 07X, FIN BASE, 07X, TROUGH END, //, 7X, DEGREES C, 0
&6X, DEGREES C, 06X, DEGREES C, 06X, DEGREES C, 05X, DEGREES C, 06X
&, DEGREES C, //)
395 FORMAT (//, 17X, MAJOR TEMPERATURES PER UNIT SECTION FOR ALL INCREME
&NTS, //, 7X, EXTERNAL, 07X, EXTERNAL, 07X, EXTERNAL, 07X, INTERNAL,
&6X, INTERNAL, 07X, INTERNAL, 07X, LEFT, 08X, MIDDLE, 08X, RIGHT
&, 09X, LEFT, 07X, MIDDLE, 07X, RIGHT, //, 7X, DEGREES F, 0
&6X, DEGREES C, 06X, DEGREES C, 06X, DEGREES C, 05X, DEGREES C, 06X
&, DEGREES C, //)
400 FORMAT (1X, I2, 4(5X, G10.5), 2(4X, G10.5))
410 FORMAT (1H0, //, 10X, 63HNODAL POINT COORDINATES AND TEMPERATURE AT
1A SPECIFIC INCREMENT, //, 10X, 58HHEAT TRANSFER COEFFICIENT, ELEMENT LE
2NGTH AND HEAT RATE AT, //, 10X, 55HCONVECTIVE BOUNDARY ELEMENTS AT TH
3IS SPECIFIC INCREMENT, ///)
420 FORMAT (1H0, 10X, 17HINCREMENT NUMBER=, I5, //)
430 FORMAT (1H0, 39HNODAL PCINT COORDINATES AND TEMPERATURE, //, 1X, 11HNOD
1AL POINT, 5X, 8HX-COORD, 09X, 7HY, 11HTEMPERATURE, //, 17X, 6HMETE
2RS, 11X, 6HMETERS, 11X, 9HDEGREES C, //)
440 FORMAT (1X, I5, 4X, 3(07X, G10.5))
450 FORMAT (1H0, 10X, 38HCONVECTIVE BOUNDARY ELEMENT PARAMETERS, //, 01X, 11
1HELEMENT NR, 10X, 6HLENGTH, 10X, 13HHEAT TRANSFER, 04X, 25HHEAT RATE PE
2R UNIT LENGTH, //, 22X, 6HMETERS, 11X, 11HCOEFFICIENT, 12X, 11HWATTS/METER,
3/38X, 14HWATTS/M2-DEG K, //)
460 FORMAT (01X, I5, 3(12X, G10.5))
END
C
C
SUBROUTINE CELCRV(NDIV, NI, ITER, IFLUID, CL1, TSAT, TAVERG, KBASE1, OMEGA
&, DELMAX, DELSTR, DRVTE, TAVG, NDEL, TALFA, CALFA, ZZERO, BFINI, EPS, T, CRIT
&, T1, TBSFIN, NTERM)
IMPLICIT REAL*8(A-H, O-Z)
DIMENSION ICOR(110, 2), DELS AV(110), GK(110, 110), DEL1(110)
DIMENSION GF(110, 1), EK(4, 4), EB(4, 4), DERIV(110), NDCF(110, 4)
DIMENSION ELNGTH(110), DELSTR(110), WKAREA(13000)
DIMENSION CNST(100), TAVG(100), DRV SAV(110), EPS(100), DERIV1(110)
DIMENSION CNST1(100), CNST2(100), T(100), EC(4, 4), T1(100), TBSFIN(100)
COMMON/DELL/RELAX
BFTMP=BFINI
IF(NDEL.EQ.0) GO TO 10
NF IN=1
GO TO 30
IF(ITER.EQ.1) GO TO 20
NF IN=1
GO TO 30

```



```

20  CONTINUE
    NF IN=0
30  CONTINUE
    IF (NDEL.EQ.0) BFIN=0.000
    IF (NDEL.EQ.1) BFIN=BFIN1
    IF (ITER.GT.1) BFIN=BFIN1
    C    WRITE(6,*) NCALL, NI, NDEL, NDEL FN, DEL MAX
    C
    NCALL=NCALL+1
    NCOUNT=1
    C
    R=RBASEI/12.000
    CL=CL1/12.000
    DELX=CL/DFLCAT (NDIV)
    NEL=NDIV+1
    EPSO=EPS(NI)
    C
    TAVGIM=TAVERG
    DO 100 I=1, NEL
    IF (NDEL.EQ.0) C=AND.ITER.EQ.1) GO TO 60
    IF (I.EQ.1) GC TO 40
    TAVERG=(TAVG(I)+TAVG(I-1))/2.000
    TWALL=(T1(I)+T1(I-1)+TBSFIN(I)+TBSFIN(I-1))/4.000
    GO TO 60
    IF (I.EC.NEL) GO TO 50
    TAVERG=TAVG(I)
    TWALL=(T1(I)+TBSFIN(I))/2.000
    GO TO 60
    CONTINUE
    TAVERG=TAVG(I-1)
    TWALL=(T1(I-1)+TBSFIN(I-1))/2.000
    CONTINUE
    TO=TAVERG
    TFILM=(TAVERG+TSAT)/2.000
    C    WATER PROPERTIES
    C
    IF (IFLUID.EQ.1) GO TO 70
    HFG=1093.8800-0.570300*TSAT+0.0001281900*(TSAT**2)-0.000000862400*
    1(TSAT**3)
    RHOF=62.77400-0.0025569800*TFILM-0.00005557200*TFILM**2
    CF=0.303400+0.0073892700*TFILM-0.000014732100*TFILM**2
    UF=(0.000139700-C.00001466900*TFILM+0.00000063125300*TFILM**2-
    1.00000000057656900*TFILM**3)*3600.000
    CP=-0.0000000000700*TFILM**3+0.000001476400*TFILM**2-0.0000276
    88000*TFILM+1.010911700
    C
    C    FRECN PROPERTIES
    C

```



```

C      DEFINE LENGTH OF ELEMENTS
      ELNGTH(1)=DELX/2.0D0
      ELNGTH(NEL)=ELNGTH(1)
      NEL1=NEL-1
      DO 140 I=2,NEL1
        ELNGTH(I)=DELX
      CONTINUE
140  C
    C
    C
    C
    C
150  C
      INITIALIZE TROUGH THICKNESS(DEL) AND DERIVATIVE (DERIV)
      AT EACH NODAL POINT
      CONTINUE
      DEL(1)=DELMAX
      NSNP1=NSNP-1
      DO 160 NP=2,NSNP1
        CEL(NP)=CEL(NP-1)-(0.6D0*DELMAX/DFLOAT(NCIV))
        DERIV(NP)=0.0D0
      CONTINUE
      IF(NFIN.EQ.C)GO TO 170
      DO 170 NP=1,NSNP
        DEL1(NP)=DELMAX
        DERIV1(NP)=DERIV(NP)
      CONTINUE
170  C
    C
    C
    C
180  C
      INITIALIZE K AND F MATRIX
      DO 200 I=1,NDOFT
        DO 190 J=1,NDOFT
          GK(I,J)=0.0D0
        CONTINUE
        GF(I,1)=C.0D0
      CONTINUE
190  C
    C
    C
    C
      FORM GLOEAL K AND F MATRICES
      JJ=0
      DO 270 IEL=1,NEL
        II=2*IEL-1
        JJ=II+1
        MM=JJJ+1
        NN=MM+1
        I1=ICGR( IEL,1)
        I2=ICGR( IEL,2)
        CELAVG=(CEL(I2)+DEL(I1))/2.0D0

```


210

```

DRVAVG=(DERIV(I 2)+DERIV(I 1))/2.000
IF(NFIN.EQ.0)GO TO 210
DELAvg=(CEL(I 2)+DEL(I 1))/2.000
DRVAVG=(DERIV(I 2)+DERIV(I 1))/2.000
CONTINUE
ZSTAR=ZZERO-(DELAvg/CALFA)
EK(1,1)=-6.000/(5.000*ELNGTH(IEL))
EK(1,2)=-11.000/10.000
EK(1,3)=-1.000*EK(1,1)
EK(1,4)=-1.000/10.000
EK(2,1)=EK(1,4)
EK(2,2)=-2.000*ELNGTH(IEL)/15.000
EK(2,3)=-1.000*EK(2,1)
EK(2,4)=ELNGTH(IEL)/30.000
EK(3,1)=EK(1,3)
EK(3,2)=EK(2,3)
EK(3,3)=EK(1,1)
EK(3,4)=-1.000*EK(1,2)
EK(4,1)=EK(2,1)
EK(4,2)=EK(2,4)
EK(4,3)=EK(3,2)
EK(4,4)=EK(2,2)
EB(1,1)=-1.000/2.000
EB(1,2)=ELNGTH(IEL)/10.000
EB(1,3)=-1.000*EB(1,1)
EB(1,4)=-1.000*EB(1,2)
EB(2,1)=EB(1,4)
EB(2,2)=EB(1,2)
EB(2,3)=EB(1,2)*ELNGTH(IEL)**2/60.000
EB(2,4)=(-1.000*EB(1,1)
EB(3,1)=EB(2,1)
EB(3,2)=EB(2,1)
EB(3,3)=EB(1,3)
EB(3,4)=EB(1,2)
EB(4,1)=EB(3,4)
EB(4,2)=-1.000*EB(2,4)
EB(4,3)=EB(3,2)
EB(4,4)=EB(2,2)
EB(4,4)=EB(2,2)
IF(NFIN.EQ.0)GO TO 220
CDEL=DABS(CNST2(IEL)*ZSTAR)**.75*CALFA
IF(CALFA.EQ.1.000)AND(BFIN1.NE.0.000)CDEL=CDEL/EPST
CNST(IEL)=CNST1(IEL)-2.000*CDEL*DELAvg
CB=(3.000*EPST*CDELAVG**3*DRVAVG)+(4.000*CDELAVG**4*ALFA*DRVAVG)
CONTINUE
EF(1)=(CNST(IEL)*ELNGTH(IEL))/2.000)
EF(2)=(CNST(IEL)*ELNGTH(IEL)**2)/12.000
EF(3)=EF(1)

```

220


```

EF(4)=-1.0D0*EF(2)
DO 260 J=1,4
  JJ=NDOF(IEL,J)
  DO 270 K=1,4
    KK=NDOF(IEL,K)
    IF(NFIN.EQ.1.AND.CALFA.NE.1.0D0) GO TO 230
    EB(J,K)=(EK(J,K)*DELAVG**4)
    EB(2,K)=(EB(J,K)*3.0D0*DELA VG**3*DRVAVG)
    GO TO 240
  CONTINUE
  EK(J,K)=EK(J,K)*((EPSO*DELA VG**4)+(DELA VG**5*ALFA))
  EB(J,K)=EB(J,K)*CB
  CONTINUE
  GK(JJ,KK)=GK(JJ,KK)+EK(J,K)+EB(J,K)
  CCNT INUE
  CCNT INUE
  GF(11,1)=GF(11,1)+EF(1)
  GF(JJ,1)=GF(JJ,1)+EF(2)
  GF(MN,1)=GF(MN,1)+EF(3)
  GF(NN,1)=GF(NN,1)+EF(4)
  CONTINUE
  APPLY BOUNDARY CONDITIONS
  NDOFT1=NDOFT-1
  DO 280 I=1,NDOFT
    GK(1,I)=0.0D0
    GK(2,I)=0.0D0
    GK(NDOFT1,I)=0.0D0
  CONTINUE
  GK(1,1)=1.0D0
  GK(2,2)=1.0D0
  GK(NDOFT1,NDOFT1)=1.0D0
  GF(1,1)=DELMAX
  GF(NDOFT1,1)=0.25D0*DELMAX
  GF(2,1)=0.0D0

      SOLVE FOR NEW DEL
      CALL LECT2F(GK,1,NDOFT,110,GF,5,WKAREA,IER)

      SAVE OLD DEL (DELSAV) FOR CONVERGENCE TEST AND DEFINE
      SOLUTION VECTOR AS DEL AND DERIV
      NP1=1
      DO 290 NP=1,NSNP
        DELSAV(NP)=DEL(NP)
        DRVSAV(NP)=DERIV(NP)

```


LIST OF REFERENCES

1. Ballback, L. J., The Operation of a Rotating Wickless Heat Pipe, M. S. Thesis, Naval Postgraduate School, Monterey, California, December 1969.
2. Tantrakul, C., Condensation Heat Transfer Inside Rotating Heat Pipe, M. S. and M. E. Thesis, Naval Postgraduate School, Monterey, California, June 1977.
3. Schafer, C. E., Augmenting the Heat Transfer Performance of Rotating Two-Phase Thermosyphons, M. S. Thesis, Naval Postgraduate School, Monterey, California, December 1972.
4. Corley, R. D., Heat Transfer Analysis of a Rotating Heat Pipe Containing Internal Axial Fins, M. S. Thesis, Naval Postgraduate School, Monterey, California, June 1976.
5. Purnomo, S. P., The Enhancement of Heat Transfer in a Rotating Heat Pipe, M. S. and M. E. Thesis, Naval Postgraduate School, Monterey, California, June 1978.
6. Davis, W. A., Optimization of an Internally Finned Rotating Heat Pipe, M. S. Thesis, Naval Postgraduate School, Monterey, California, September 1980.
7. Vanderplaats, G. N., COPEs-A Fortran Control Program for Engineering Synthesis, User's Manual, prepared for a graduate course on numerical optimization presented at the Naval Postgraduate School, Monterey, California, 1979.
8. Leppert, G., and Nimmo, B. G., "Laminar Film Condensation on Surfaces Normal to Body or Inertial Forces," Transactions of the ASME, p. 178, February 1968.
9. Nimmo, B. G., and Leppert, G., Laminar Film Condensation on Finite Horizontal Surface, Clarkson College of Technology, Potsdam, New York 1970.
10. Wagenseil, L. L., Heat Transfer Performance of Various Rotating Heat Pipes, M. S. Thesis, Naval Postgraduate School, Monterey, California, December 1976.
11. Nusselt, W., "Die Oberflächenkondensation des Wasserdampfes," Zeitschrift des Vereines deutscher Ingenieure, Vol. 60, pp. 541 and 569, 1916.

12. Sparrow, E. M., and Gregg, J. L., "A Theory of Rotating Condensation," Journal of Heat Transfer, Vol. 81, Series C, pp. 113-120, May 1959.
13. Daniels, T. C., and Al-Jumaily, F. K., "Investigations of the Factors Affecting the Performance of a Rotating Heat Pipe," International Journal of Heat and Mass Transfer, Vol. 18, pp. 961-973, 1975.

INITIAL DISTRIBUTION LIST

	No. Copies
1. Defense Technical Information Center Cameron Station Alexandria, Virginia 22314	2
2. Library, Code 0142 Naval Postgraduate School Monterey, California 93940	2
3. Department Chairman, Code 69 Department of Mechanical Engineering Naval Postgraduate School Monterey, California 93940	1
4. Dr. Paul J. Marto, Code 69 Department of Mechanical Engineering Naval Postgraduate School Monterey, California 93940	2
5. Dr. David Salinas, Code 69Zc Department of Mechanical Engineering Naval Postgraduate School Monterey, California 93940	2
6. LT Adam F. Kleinholz 13734 Birkhall Avenue Bellflower, California 90706	3

201686

Thesis

K579 Kleinholz

c.1 An analysis of
smooth and axially
finned, rotating heat
pipe condensers.

19 MAR 84

27893

201686

Thesis

K579 Kleinholz

c.1 An analysis of
smooth and axially
finned, rotating heat
pipe condensers.

thesK579

An analysis of smooth and axially finned



3 2768 002 10623 9

DUDLEY KNOX LIBRARY